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BELT CONVEYORS.

By H. DAVIES.

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BELT CONVEYORS.

By H. Davies.

INTRODUCTION.

Few types of machinery have ever attained the widespread popularity for genuine usefulness that the belt conveyor experiences. The general adoption of the belt conveyor as a mechanical carrier for bulk and package materials has been entirely warranted by the inherent merits of the class of machinery. Within recent years the scope of service of the belt conveyor has widened, with the result that the users of this equipment are classed in a widely diversified list of industries.

To the mechanical engineer, practical operator or mechanical handling draughtsman, the principal desirable factors which characterize the belt conveyor in general are apparent.

Simplicity of construction with few parts comprising the whole system, thus being low in initial cost in rates to capacity.
High efficiency and unrivalled dependability.
Positive action and non-mixing features.
Complete discharge of the material being handled.
Small driving power required and economy of upkeep.
Wide range of adaptability.

The mechanical movement of raw materials and goods by belt conveyors has revolutionized practically every industry and accelerated progress in the production methods of most manufacturers and distributors. Saving of time and money has been effected and many plants and factories are now designed upon and around belt conveyor systems.

FEATURES.

One great feature of the belt conveyor is the perfect separation of the conveying parts from the running parts. The material is received directly on to the belt and is carried with a minimum of friction to its destination. There is none of the jamming and clogging so noticeable in the operation of the scraper systems: nor are there any joints or other projections to break or wear. The point where the load is received is the only point of abrasion or friction between the material and the belt. The advantage in this respect over any form of flight conveyor, in which the friction is constant all along the haul, is very obvious. Another great point is that the conveyor is practically noiseless when running.

USES.

Belt conveyors have practically an unlimited field where material is required to be transported quickly and cheaply from one point to another. Materials of all descriptions, whether wet or dry, both light and heavy, can be successfully handled in practically any quantity. In addition to handling loose bulk materials, such as coal, coke, sand, gravel, sugar beet, etc., extensive use is now made of the belt conveyor for handling general merchandise, such as boxes, crates, etc., in warehouses, canneries and factories, newspapers in printing offices and letters and parcels in post offices, etc.

CONVEYOR BELTING.

British Standard Specification 490, 1933 and 1951. The following extracts indicate the minimum standards of quality and performance set up for both conveyor and elevator belts by the Mechanical Industry Committee of the British Standard Institute, which comprises representatives of H.M. Government departments, the principal engineering institutes and associations, railway companies, etc.

The three grades of belting provided for are as follows :—

Grade "A" (Severe conditions).
,, "B" (Average ,,)
,, "C" (Light ,,)

The following table, No. 1, is given as a guide to selection of the grade of belt to adopt.

These are, of course, very broad classifications and constitute a guide only to the grade of belt required.

Rubber belting manufacturers are always willing to advise upon the most suitable type of belt to use to suit any type of material to be handled.

Duck.

The conveying belt itself can lay claim to the greatest consideration in the whole of the device. It should have the correct amount of flexibility to enable it correctly to wrap round the end pulleys and also be of sufficient width to carry the required amount of material. In addition to these two factors there is also the weight of the material on the belt and the pull exerted at the driving pulley to drive the conveyor to consider, and the belt should be of sufficient strength to withstand these loads.

Rubber and canvas belts are made up of a number of plies of cotton duck cemented together by an elastic rubber compound. The strength of the belt is confined to the cotton duck and this should be of the most economical combination of weight and strength it is possible to secure.

TABLE No. 1.

Type of Material.	*Examples.*	*Grade of Belt.*
Non-Abrasive and dry	Flour, powders, wood chips, pulp, boxes, bales, bags, sacks, paper.	C
Very slightly Abrasive, or wet	Soft sand, soda, earth, clay, cement, charcoal, sugar, grain, etc.	B or C
Slight Abrasive	Small coal, fine gravel, rubble, sharp sand, crushed bones	B
Moderately Abrasive ...	Coke under 2″ Coal, coarse gravel, ashes.	B
Sharp or Abrasive ...	Crushed ores, Sandstone, stone, chippings, slate, coke over 2″, broken glass.	A
Very severe conditions Hot materials over 150°F. up to 230°F.		A Heat resisting A

The standard measurement of the weight of duck is given in ounces for a piece 36″ long by 42″ wide, and standard conveyor duck is 28 ounces for this measurement.

Most belt manufacturers also make belts having 32 ozs. and 36 ozs. and even 42 ozs. duck.

Friction.

The cementing layer of friction or the rubber which holds the plies together is applied to the duck in a plastic state by passing between two rolls, running at slightly different speeds, which drive this rubber completely through the duck and impregnate it thoroughly, thereby surrounding each fibre with rubber. This process causes the rubber to permeate the whole fabric and makes it impervious to moisture and other deleterious agencies. It also serves as a preservative and lubricant against internal wear or friction of the threads.

The friction strength is measured by the pull required to separate one ply from its neighbour at the rate of a 1″ wide strip at 1″ per minute.

Cover.

After the plies are laid together the rubber cover is put on the outside. This rubber covering is really the wearing cover to protect the cotton duck from abrasion and cutting and to exclude any moisture that may be present. The top cover in thickness and grade is proportioned to the abrasiveness of the material to be carried.

For fine materials, such as grain, corn, sugar, sawdust, shavings, clay, etc., the cover varies from $\frac{1}{32}''$ to $\frac{1}{16}''$ dependent on the abrasiveness of the material. Crushed coal, small stone and screened materials do not cause much abrasion, and $\frac{1}{16}''$ to $\frac{1}{8}''$ cover is generally adopted. For heavy ores, lump stone or rock, 2″ to 6″ cube, run of mine coal and all material of a similar heavy or cutting nature $\frac{3}{16}''$ thick cover is advisable, particularly for short belts.

The back cover is normally about $\frac{1}{32}''$ to $\frac{1}{16}''$ thick, according to conditions, and is required chiefly to insulate the back of the belt from deteriorating agents and to provide for wear against slipping on the pulleys, particularly when operating under gritty or dusty conditions.

Clause 5 (*a*) of British Standard Specification provides that the tensile strength and elongation of rubber covers shall attain the following minimum values under test, as given in Table No. 2.

TABLE No. 2.

Grade.	Ultimate Tensile Stress.		Elongation per cent.
	Lbs. per sq. inch.	Lbs. per m/m.	
A	3500	2·46	550
B	2500	1·76	450
C	1500	1·05	350

Clause 5 (*b*) Thickness, stipulates that the rubber covers shall not be less than $\frac{1}{32}''$ (0·8 mm.) thick on both the pulley and carrying side of the belting. On the working side, the thickness must be increased to a degree appropriate to the material to be handled and to the specific loading. Typical thicknesses recommended for belts under normal conditions in relation to grade, are as follows :—

Grade "A"	$\frac{3}{16}''$	(4·8 m.).
,, "B"	$\frac{1}{8}''$	(3·2 m.).
,, "C"	$\frac{1}{32}''$	(0·8 m.).

Strength of Fabric.

Clause 11 provides that "the average ultimate tensile strength of the fabric or cotton duck of the finished belting shall be not less than the following values, as given in Table No. 3.

TABLE No. 3.

Weight of Fabric.	Ultimate Tensile Strength.			
	Warp.		Weft.	
	Us/in. width.	Kg/m/m. width.	Us/in. width.	Kg/m/m. width.
28 oz. Soft	350	6·25	170	3·04
31 oz. Hard	350	6·25	200	3·57
32 oz. Soft	400	7·14	185	3·30
33½ oz. Hard	400	7·14	250	4·46
36 oz. Soft	420	7·49	200	3·57
42 oz. Soft	500	8·93	240	4·28

Strength of Belts.

While it requires a pull of about 380 lbs. per inch of width to break a single ply of belt fabric, it has been found by experience that a continued strain in excess of 25 lbs. per inch per ply not only endangers the hold of fasteners, but causes elongation of the belt and puts undue tension on the rubber cover. Since rubber cuts more easily when under tension, the resistance of the cover to injury is impaired by excessive stretch in service. Fabric under extreme tension is also more liable to ruptures from hard blows.

Table No. 4 is based upon 25 lbs. per inch per ply for standard conveyor belt duck weighing 28 ounces per yard 42″ wide. This is increased to 30 lbs. for 32 ounces duck and 32 lbs. for 36 ounce duck and 40 lbs. for 42 oz. duck.

Fabrics weighing 31 oz. and 33½ oz. are classed as "hard" being specially designed to give greater weft strength and rigidity, thus providing increased bolt holding ability when used as elevator belts.

Adhesion.

Clause 12 stipulates that adhesion between cover and plies and between individual plies shall be such that the rate of separation under the loads specified below shall not be greater than 1″ (25·4m/m) per m/m, as given in Table No. 5.

TABLE No. 4.
SAFE WORKING STRESS OF CONVEYOR BELTS.
In pounds per inch of width.

Plies.	28 oz. per yard 42″ wide.	32 oz. per yard 42″ wide.	36 oz. per yard 42″ wide.	42 oz. per yard 42″ wide.
3	75	90	96	120
4	100	120	128	160
5	125	150	160	200
6	150	180	192	240
7	175	210	224	280
8	200	240	256	320
9	225	270	288	360
10	250	300	320	400
11	275	330	352	440
12	300	360	384	480

TABLE No. 5.

Grade of Belt.	Type of Fabric.	LOAD.			
		Cover to Ply.		Ply to Ply.	
		Lbs.	Kg.	Lbs.	Kg.
A	Soft	24	10·9	26	11·8
A	Hard	20	9·1	23	10·4
B	Soft	18	8·2	20	9·1
B	Hard	16	7·3	18	8·2
C	Soft	15	6·8	17	7·7
C	Hard	14	6·4	16	7·3

Note.—Where the thickness of the rubber cover is less than $\frac{1}{16}$″ (1·59 m/m) but not less than $\frac{1}{32}$″ (0·8 m/m) a tolerance of minus 20% shall be permitted on the "cover to ply" adhesion figures given in the foregoing table.

Standard tests are laid down in the specification for each of the foregoing properties and characteristics.

Stepped Ply Belts.

Where the conditions of loading cause the wear to concentrate on the central position of the belt, the protective layer of rubber is consequently much more worn in this part of the belt than on the outer edges and the adoption of a stepped ply belt or a straight ply belt having a thicker cover in the centre should be considered. The stepped ply belt is now gradually losing favour in place of the straight ply belt, although such belts conform more readily to troughing and have the distinction of providing additional rubber cover where the wear is most severe.

Edge.

A perfect edge is obtained by taking the top cover and carrying it in slightly diminished thickness tightly round the edge and well into the back cover. This provides the maximum amount of rubber possible and anchors it on this lower dangerous corner into the body of the rubber in the back of the belt. This form of edge, even in diminished thickness, provides normally sufficient wear resistance to last the life of the belt and will adhere until it is worn away.

Manufacture.

The construction and manufacture of a belt, while calling for great experience and perfection, do not necessarily determine a good belt unless all its parts have been carefully proportioned so as to combine the finished belt into a unified product, every part of which is properly proportioned and adapted to the service required of it. All conveyor belts should have every part proportioned economically to the requirements of that part and so proportioned to each other that they will all give their service without undue waste or extravagance in any part.

Temperature.

Rubber and canvas conveyor belts are now manufactured for carrying hot materials, such as coke, sand and cement clinker. The maximum temperature should not exceed 230° Fah.

Weights.

The weights given in Table No. 7 are based on Table No. 6.

For each $\frac{1}{32}''$ of cover thickness add ·0/8 lbs. per inch of width.

Belt Tensions.

The maximum working tensions in the belt should not exceed the following, as given in Table No. 8.

Types of Rubber Belts.

There are many special types of rubber covered conveyor belts, such as those specially treated to resist oil, others to withstand heat and others with rubber covers which are tasteless and odourless for the conveyance of foodstuffs.

Balata Belts for conveyors do not seem to find great favour and are rapidly being displaced by the rubber and canvas belt. They find their chief application for power transmission belts and also for belt elevators handling wet materials, due to their great strength and freedom from stretch. Their waterproof quality is a great advantage, particularly for handling wet materials, such as mineral pulps, but owing to their stiffness and density they are usually only adopted on flat carriers. They can be used on troughing idlers, particularly if the angle of same is not excessive, but are usually limited to the flat pulley type.

The duck, as used for conveying belts, is of a much closer formation than that used for rubber belts and weighs from 38 to 40 ozs. per square yard. The cotton duck, which is first waterproofed, is held together by balata, a true gum which is stronger than rubber but not so elastic. Since the fibres of the duck are impregnated with this gum it offers a great resistance to water, but due to the gum softening at a rather low temperature, about 120° Fah., the belt must be protected against heat.

Canvas Belts.

Canvas belts, as used for both conveyors and elevators, are made of heavier duck than the rubber and canvas belts, being more closely woven. The weight is also taken on a different basis, the cotton duck being usually rated at 32 ozs. per square yard as against the 36″ × 42″ for the rubber belt. The construction of canvas belts varies with different makers, but practically all are made on the warp and filler principle.

After the stitching of the belts, they are subjected to pressing and stretching before being treated with a waterproof compound. The belt is then pressed between rolls to squeeze out the surplus compound.

The basis of the usual waterproofing compound for ordinary conveyor and elevator belts is linseed oil, which, being thin, penetrates all the fibres of the cotton forming the duck. The oil remaining after the belt has been rolled forms a waterproofing and also helps to make the belt more pliable to the internal bending which takes place.

This process also adds to the wearing quality of the belt, the threads being much tougher, due to the penetration of the oil compound. After the belt is finally dired and cured by stretching, a coat of paint made of a mineral pigment and a drying oil is applied, which fills up the stitching holes and produces a hard, tough wearing surface. The outer coating prevents the oil compound from working out when the belt is in operation and further gives the belt a good coefficient of friction for contact with the driving pulley.

Although these belts are waterproofed as previously stated, they do not prove suitable for operating under wet conditions. They are also unsuitable under great heat and do not withstand chemical fumes or acids.

Solid Woven Belts.

These belts differ from the canvas and balata belt, being loom woven with a warp and weft their full width, and not made up of plies. They are usually sold under their trade names by the various makers and really are intended for power transmission belts. Use is made of these belts for conveying purposes, but only for handling light packages in buildings. They are apt to stretch more than the canvas and rubber belts and are therefore limited to a small tensile pull. In some instances they are treated with different kinds of waterproofing compounds and for handling granite a special bitumenized solid woven cotton belt has proved very efficient.

Plasticon conveyor belting has been specially designed by the makers, Mulcott Belting Co., Ltd., Bradford, to meet the growing demand for a conveyor belt completely hygienic, odourless and impervious to moisture, oils and acids. It is particularly suitable for conveying food and largely used in food processing industries, such as for handling chocolates, biscuits, toffees, margarine, fruits, meat, bread and cakes, vegetables, tea, etc., etc. Plasticon can be cleaned easily and quickly with hot water or steam, the surface will not peel or crack and will work satisfactorily in temperatures ranging from 0°C. to 100°C.

TABLE No. 6.

Lbs. per inch per Ply.	Weight of Duck.
·021	28
·024	32
·026	36
·029	42

TABLE No. 7.

Belt Width.	No. of Plies.	28 Ounce Duck.							32 Ounce Duck.								36 Ounce Duck.					42 Ounce Duck.				
		Rubber Cover Thickness Carrying Side.							Rubber Cover Thickness Carrying Side.								Rubber Cover Thickness Carrying Side.					Rubber Cover Thickness Carrying Side.				
		1/16	1/8	3/16	1/8	3/16	1/4	1/4	1/16	1/8	3/16	1/8	3/16	1/4	1/4	3/8	1/8	3/16	1/4	1/4	3/8	1/8	3/16	1/4	1/4	3/8
		Rubber Cover Thickness Pulley Side.							Rubber Cover Thickness Pulley Side.								Rubber Cover Thickness Pulley Side.					Rubber Cover Thickness Pulley Side.				
		1/32	1/32	1/32	1/16	1/16	1/16	1/8	1/32	1/32	1/32	1/16	1/16	1/16	1/8	1/8	1/16	1/16	1/16	1/8	1/8	1/16	1/16	1/16	1/8	1/8
12	3	1·4	1·8	2·3	2·1	2·5			1·5	1·9	2·4	2·2	2·6													
	4	1·7	2·1	2·5	2·3	2·7			1·8	2·2	2·7	2·5	2·9													
14	3	1·6	2·1	2·6	2·4	2·9			1·3	2·3	2·8	2·5	3·0													
	4	1·9	2·4	2·9	2·7	3·2			1·6	2·6	3·1	2·9	3·4													
	5	2·2	2·7	3·7	3·0	3·5			1·9	2·9	3·4	3·2	3·7													
16	3	1·9	2·5	3·0	2·7	3·3			2·0	2·6	3·2	2·9	3·5													
	4	2·2	2·8	3·4	3·1	3·7			2·4	3·0	3·6	3·3	3·8													
	5	2·5	3·1	3·7	3·4	4·0			2·8	3·4	3·9	3·6	4·2													
18	4	2·5	3·1	3·8	3·5	4·1			2·7	3·3	4·0	3·7	4·3													
	5	2·9	3·5	4·2	3·8	4·5			3·1	3·8	4·4	4·1	4·8													
	6	3·2	3·9	4·5	4·2	4·9			3·6	4·2	4·9	4·5	5·2													
20	4	2·8	3·5	4·2	3·8	4·6			3·0	3·7	4·4	4·1	4·8													
	5	3·2	3·9	4·6	4·3	5·0			3·5	4·2	4·9	4·6	5·3													
	6	3·6	4·3	5·0	4·7	5·4			4·0	4·7	5·4	5·1	5·8													
24	4		4·2	5·0	4·6	5·5	6·3			4·5	5·4	4·9	5·8	6·6			5·1	6·0	6·8	7·7	9·4	5·4	6·2	7·1	7·9	·97
	5		4·7	5·5	5·1	6·0	6·8			5·0	5·9	5·5	6·3	7·2			5·7	6·6	7·4	8·3	10·0	6·1	6·9	7·8	8·6	10·4
	6		5·2	6·0	5·6	6·5	7·4			5·6	6·5	6·0	6·9	7·8			6·3	7·2	8·1	8·9	10·7	6·8	7·6	8·5	9·3	11·1

TABLE No. 7—continued.

Belt Width.	No. of Plies.	28 Ounce Duck.							32 Ounce Duck.								36 Ounce Duck.					42 Ounce Duck.				
		Rubber Cover Thickness Carrying Side.							Rubber Cover Thickness Carrying Side.								Rubber Cover Thickness Carrying Side.					Rubber Cover Thickness Carrying Side.				
		1/16	1/8	3/16	1/8	3/16	1/4	1/4	1/16	1/8	3/16	1/8	3/16	1/4	1/4	3/8	1/8	3/16	1/4	1/4	3/8	1/8	3/16	1/4	1/4	3/8
		Rubber Cover Thickness Pulley Side.							Rubber Cover Thickness Pulley Side.								Rubber Cover Thickness Pulley Side.					Rubber Cover Thickness Pulley Side.				
		1/32	1/32	1/32	1/16	1/16	1/16	1/8	1/32	1/32	1/32	1/16	1/16	1/16	1/8	1/8	1/16	1/16	1/16	1/8	1/8	1/16	1/16	1/16	1/8	1/8
30	5		5·9	6·9	6·4	7·5	8·6			6·3	7·4	6·8	7·9	9·0			6·7	8·3	9·4	10·4	12·6	7·6	8·7	9·8	10·8	13·0
	6		6·5	7·6	7·0	8·1	9·2			7·0	8·1	7·6	8·7	9·7			7·4	9·1	10·1	11·2	13·3	8·5	9·5	10·6	1·7	13·9
	7		7·1	8·2	7·7	8·7	9·8			7·7	8·8	8·3	8·4	10·5			8·2	9·9	10·9	12·0	14·2	9·3	10·4	11·5	12·6	14·7
36	6		7·8	9·1	8·4	9·7	11·0	12·3		8·4	9·7	9·1	10·4	11·7	13·0		9·5	11·4	12·1	13·4	16·0	10·2	11·4	12·7	14·0	16·6
	7		8·5	9·8	9·2	10·5	11·8	13·1		9·3	10·6	10·0	11·2	12·5	13·8		10·4	12·4	13·0	14·3	16·9	11·2	12·5	13·8	15·1	17·7
	8		9·3	10·6	10·0	11·2	12·5	13·8		10·2	11·5	10·8	12·1	13·4	14·7		11·4	13·3	14·0	15·3	17·9	12·2	13·5	14·8	16·1	18·7
42	6		8·9	10·4	9·7	11·5	13·0	13·9				10·6	12·1	13·6	15·1	18·2										
	7		9·9	11·4	10·7	12·3	13·8	15·0				11·6	13·1	14·6	16·1	19·2	12·2	13·7	15·2	16·7	19·8	13·1	14·5	16·1	17·6	20·6
	8		10·9	12·4	11·7	13·1	14·6	16·1				12·6	14·1	15·6	17·1	20·2	13·3	14·8	16·3	17·8	20·9	14·3	15·8	17·3	18·8	21·8
	10											14·6	16·1	17·6	19·1	22·1	15·5	16·9	18·5	20·0	23·0	16·7	18·2	19·7	21·3	24·3
48	7		11·3	13·0	12·2	14·1	15·8	16·9				13·3	15·0	16·7	18·4	21·9	13·9	15·6	17·4	19·1	22·6	14·9	16·7	17·3	18·8	21·8
	8		12·5	14·2	13·4	15·0	16·8	18·4				14·4	16·1	17·9	19·6	23·0	15·2	16·9	18·7	20·4	23·9	16·3	18·0	18·7	20·2	23·2
	10											16·6	18·3	20·1	22·0	25·3	17·7	19·4	21·1	22·9	25·3	19·1	20·8	21·5	23·0	26·0
54	8											16·2	18·2	20·0	22·0	25·9	17·0	19·0	21·0	22·9	26·8	18·4	20·3	22·2	24·2	28·0
	10											18·8	21·0	23·0	24·5	28·5	19·9	21·8	23·8	25·7	29·6	21·5	23·4	25·4	27·3	31·2
	12											21·3	23·3	25·2	27·2	31·1	22·7	24·6	26·6	28·5	32·4	24·6	26·6	28·5	30·5	34·3
60	8											18·0	20·0	22·0	24·5	28·8	19·0	21·1	23·3	25·4	29·8	20·3	22·5	24·7	26·8	31·2
	10											21·0	23·0	25·0	27·5	31·7	22·0	24·2	26·4	28·5	32·9	23·8	26·0	28·2	30·3	34·7
	12											24·0	26·0	28·0	30·0	34·5	25·2	27·4	29·5	31·6	36·0	27·4	29·5	31·7	33·8	38·2

TABLE No. 8.

Weight of Fabric ozs.	Maximum Working Tension lb. per inch width per ply (including initial tension).
28	25
31	25
32	30
33½	30
36	32
42	40

BELT FASTENERS.

The best type of conveying belt fastener should be strong, but of a perfectly flexible character, not only in the direction of travel, but also transversely, in order to allow the joint to yield evenly and easily to the contour of the troughing idlers. The ordinary laced fasteners, as used for transmission belts, do not function well with conveyor belts, as they are apt to leave a gap between the ends of the belt and thus allow fine material to fall through. Another fault is that they transmit the pull to the cross threads of the cotton duck and if near the end are liable to pull out. Due to these defects the use of laced belts has been discarded.

Alligator.

A popular form of flexible steel lacing is the "*Alligator*" type. This is fabricated from special cold rolled steel strip having a series of prongs which are driven into the belt and clinched over.

These are made in the standard pitch which interlaces together and are connected with a double rocker pin, or a flexible steel wire and form a hinge which readily conforms to crowned pulleys. This type of fastener lies close to the belt and is admirably adapted for conveyors for handling packages.

Bristol.

The "*Bristol Belt Lacer*" is a metallic type of fastener with prongs arranged staggered, which, when driven into the belt, grips the warp threads of the duck tightly and so obtains a good firm hold.

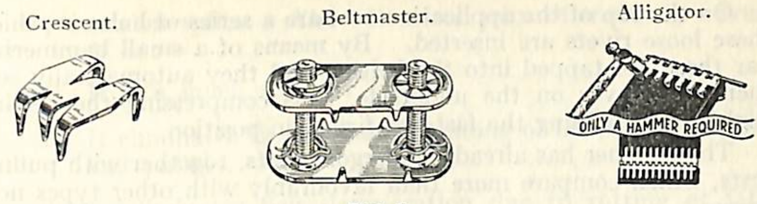

Crescent. Beltmaster. Alligator.

Fig. 1.

Crescent.

Another metallic fastener of a similar type is the *"Crescent"* lacer, which has the same advantage as the *"Bristol"* lacer, that is, if the belt becomes worn at the edges of the lacing it is a simple matter to cut it out and insert a new row. The *"Crescent"* form of fastener is usually adopted for heavy work or large size belts, and consists of a steel plate attached to the belt by means of split rivets. These rivets can be driven through the belt to compress and clinch the duck in small belts, but for large belts the holes are usually punched out.

Belt Master and H.D.

Two very popular types of conveyor fasteners are the *"Beltmaster"* and the *"H.D"* or *"Heavy Drive"* as manufactured by Messrs. Steelace Manufacturing Co. and Messrs. J. B. Stone & Co., Ltd., respectively.

The strength of the finished joint is made up on the principle of compression. The strain of the load on the conveyor is distributed over the area covered by the plates and is not taken by the bolts alone. The joint is flexible and will readily trough to conform with the angle of the idlers and can be used on crowned face pulleys. A tight butt joint is made, preventing loss or spillage of material.

New Type.

The *"Regis"* Colliery Conveyor Belt Fastener is entirely different from the orthodox method of fastening conveyor belting, which usually employs hooks for fixing or nuts and bolts, as a special hollow rivet is employed in clamping the fastener to the belt.

At the moment the *"Regis"* Colliery Conveyor Fastener is designed for the quick fixing of fasteners of 24″ and 30″ wide conveyor belting. To apply these fasteners quickly and efficiently an application tool of extremely simple design has to be employed. The "U" shaped member of the fastener is placed in the application tool. This is then clamped on to the belt by two thumb screws. Once the application tool has been tightened down it does not matter whether the belting moves or not, a faulty joint is impossible.

On the top of the application tool are a series of holes in which these loose rivets are inserted. By means of a small hammering bar these are tapped into the belting and they automatically seal themselves over on the underside, thus compressing the belting slightly and holding the fastener firmly in position.

The fastener has already undergone tests, together with pulling tests, which compare more than favourably with other types now in use.

The application tool weighs less than a tenth of the normal machine required to insert hook type fasteners.

With the aid of one application tool a joint can be made fool proof in 15 minutes, but where speed is essential in the mending of a belt, with two persons each using an application tool the belt can be completed in approximately seven minutes.

A good feature with this type of fastener is that the ends of the belt are extremely strong and the fastener does not readily pull out owing to its great strength.

Size.

The size of the metallic fasteners to use should be left to the makers of the belt. Rules are given for the various sizes or box numbers of fasteners for belts of different plies, but as previously stated, let the belt maker fix the size, and further, follow his instructions as to how to fix same.

Metal fasteners do not form a joint of equal strength to the belt and therefore care should be taken to procure a good splice. A good coating of rubber cement, particularly for conveyors handling wet or moist materials should be applied over the butt ends of the belt before making the joint. This will, to a great degree, prevent moisture from entering the duck, this precaution being particularly necessary with canvas belts.

The use of endless belts is chiefly confined to short conveyors, due to the trouble of erecting over the end drums and carrying idlers.

Vulcanised Joints.

Probably no improvement in conveyor belt practice has done more to save maintenance, time and expense than the perfection of the field vulcanised splice.

Simple and inexpensive vulcanising equipment operated by electric current is now available and most of the conveyor belt manufacturers are prepared to send such equipment with skilled operators to any part of the country to carry out such work.

Many operators now own their own vulcanisers and any intelligent mechanic can quickly learn how to make a successful and permanent splice. The vulcanised joint offers many advantages such as :—

(1) It is a much stronger joint.

(2) It eliminates the constant pounding of fasteners over idlers and pulleys.

(3) It eliminates belt deterioration due to rotting of fabric around fasteners, particularly where the conveyor is exposed to acid or water.

(4) It prevents burning out of the fabric which results when metal fasteners are exposed to hot material.

(5) It eliminates damage due to destruction of metal fasteners exposed to the abrasive action of sharp material.

(6) Material can be easily and successfully ploughed off the belt.

IDLERS.

The early forms of belt conveyors were made with the belt sliding upon a flat continuous surface, side skirt board being arranged at each side the full length of the conveyor. The return belt was also supported in the same manner. The next development was to introduce supporting rollers to eliminate the great wear experienced on the pulley side of the belt. This did to a certain degree lessen the wear, but as these conveyors were at first only used for handling light materials in flour and grain mills, they were not subjected to heavy loading.

When the handling of heavier materials, such as coal, ores and similar gritty or abrasive substances, was introduced, the wear experienced from the skirting proved a big fault and this led to the discarding of the continuous skirting.

To obtain a greater carrying capacity it was then realised that this could be obtained by giving the band a troughed form, but a fatal defect was soon discovered. It was obvious that whatever shape is given to a horizontal band the speed of the full width will be equal. Resting or being supported on a solid roller of varying diameter, it is clear that the consequent varying surface speed of the roller must set up a rubbing action as the band passes over each supporting roller set and thus result in the speedy destruction of the belt.

A distinct improvement was made when the ends of the rollers were made separate from the body, thus allowing the end rollers to operate at a slower speed than the horizontal portion of the set, to suit the belt and avoiding much destructive friction.

Three-Pulley Idlers.

Fig. 2 shows a typical three-pulley idler set. The tendency for the belt to slip and scour is eliminated as the pulleys revolve independently of each other, being mounted on separate spindles. The two outer inclined pulleys are set at varying angles, according to different makers, but 20° is now the usual angle adopted.

The plain bearing type is now practically obsolete, ball bearing, roller bearings and tapered roller bearings now being universally adopted.

Lubrication of the idlers is very important, and the various makers all put forward their own forms of grease gun pressure fittings and sealing devices.

A type of idler now becoming very prominent is the "seal for life" grease and oil packed bearings. Fig. 3 shows such an idler set.

Five-Pulley Idlers.

Fig. 4 shows a typical design of a five-pulley idler set. The outer pulleys are set at an angle of 30° and the inner pulleys at 15°, thus forming a trough of practically a curved contour.

The construction is generally as described for the three-pulley idler set.

Latest American practice discards the five-pulley idlers in favour of the three-pulley type and idler sets to suit 72″ wide belts are listed. This design troughs the belt gradually, gives highest capacity and obviously permits greater economy of construction power, maintenance, etc., than idlers with a greater number of rollers.

Flat Belt Idlers.

These are usually made of steel tube rollers, with aluminium die cast ends and fitted with ball, roller or tapered roller end bearings suitably housed, and if of the lubricating type, seals and grease gun fittings provided, or they can be of the grease or oil packed type as previously described, and as shown in Fig. 5.

Return Idlers.

These are of similar construction to the flat belt type, with various types of supporting brackets.

Side Guide Idlers.

These are used to assist in the training of the belt to run true and in line with the carrying idlers. The customary practice is to arrange the rollers on each side of the carrying run about 10 to 15 feet from each end of the conveyor and always just before a troughing idler, as shown in Fig. 6, then at distances of 30′-0″

5"— 3 ROLL TROUGHING IDLER SETS

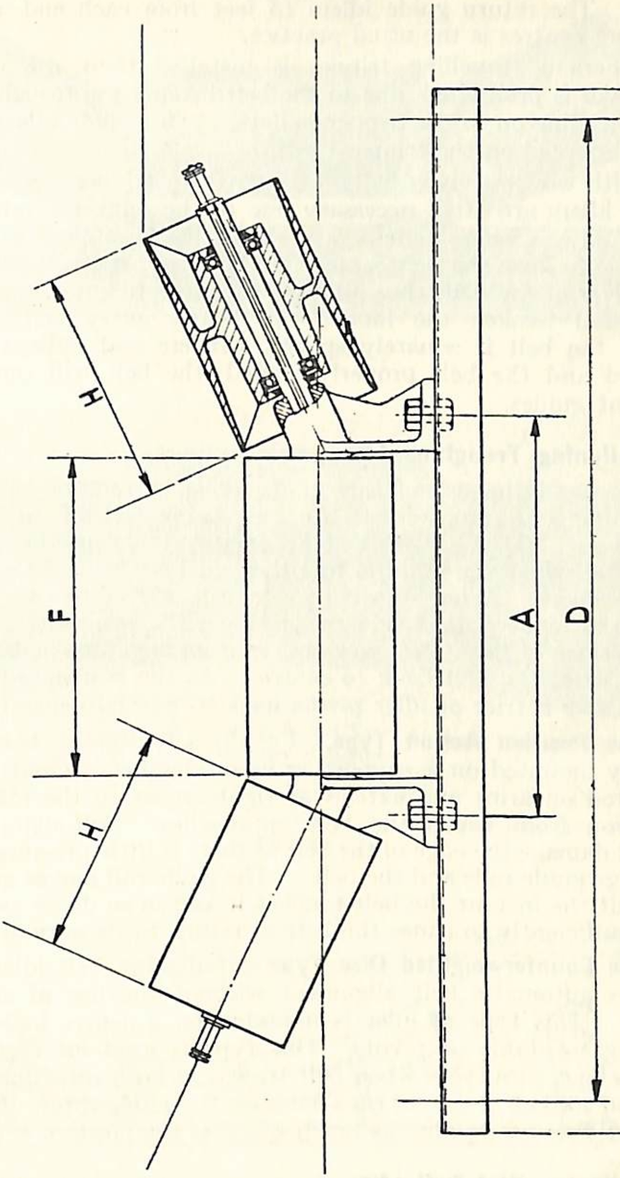


Fig. 2.

apart for 12″ to 16″ wide belts, up to 45 feet for belts 20″ wide and over. The return guide idlers 15 feet from each end and 50 to 100 feet centres is the usual practice.

Where a travelling tripper is installed their use along the conveyor is prohibited, due to the belt leaving the troughing idlers when leading on to the tripper pulleys. Side guide idlers are very often adopted on the tripper itself.

With woven canvas belts, except where trippers are used, side guide idlers are often necessary due to the uneven stretch of the belt fabric. Side guide idlers should be used as sparingly as possible to keep the belt centred. Continued use is likely to wear the edges of the belt, thus allowing moisture to enter, separate the plies and weaken the fabric. In nearly every belt conveyor, where the belt is squarely spliced, carriers and pulleys properly aligned and the belt properly loaded, the belt will centre itself without guides.

Self-Aligning Troughing Idlers.

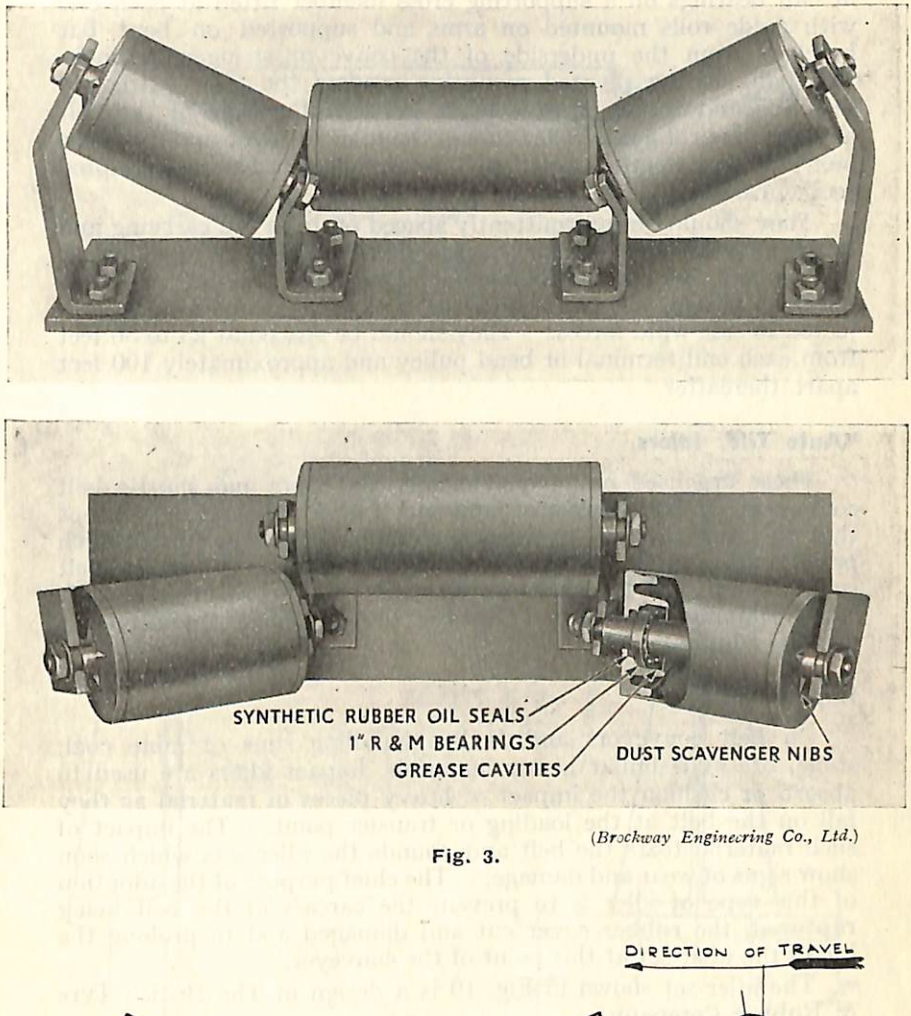
To avoid the use of side guide idlers, automatic self-aligning troughing and return idlers are now being largely used on long conveyors. These idlers will automatically bring the belt back to centre whenever it shifts to either side, without the use of side guide rollers. When the conveyor belt strays to one side, the increased load on that side swings the self-aligning idler forward. The surface of the rollers now acting at an angle to the belt travel, tend to run the belt back to centre. As the belt approaches the centre, the carrier or idler pivots back to normal neutral position.

The Positive Action Type. Fig. 7, self-aligning belt idler is pivotly mounted on a supporting cross member. Guide rolls are mounted on arms and extend at right angles to the idler in the direction from which the belt approaches. Self-aligning idlers do not damage the edge of the belt as there is little pressure between the edge guide rolls and the belt. The guide roll moves away from the belt the instant the belt touches it and in so doing swivels the idler sufficiently to cause the belt to return to its normal position.

The Counterweighted Disc Type self-aligning belt idler, Fig. 8, assures automatic belt alignment without the use of side guide idlers. This type of idler is mounted on a centre ball or roller bearing turntable or pivot. This type is used for reversible or shuttle belt conveyors when belt travels in both directions. They are also used on conveyor runs having a travelling throw off carriage since it does not require as much space as the positive action type.

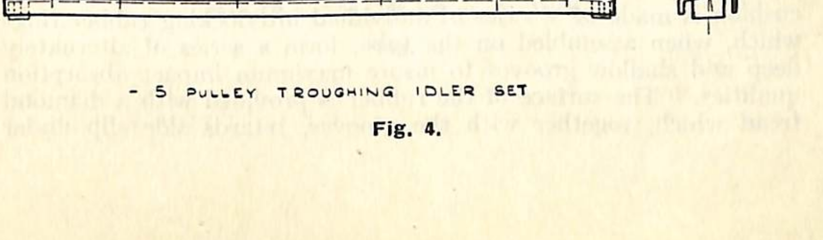
Self-Aligning Flat Belt Idler.

Fig. 9 employs the same operating principle as the troughing idlers. They consist of a standard return idler pivotly mounted



(Brockway Engineering Co., Ltd.)

Fig. 3.



- 5 PULLEY TROUGHING IDLER SET

Fig. 4.

in ball bearings on a supporting cross member fitted at each end with guide rolls mounted on arms and supported on bent bar brackets from the underside of the conveyor stringer channels. The ball bearing pivoted mounting renders the idler extremely sensitive and the slightest contact between the edge of the belt and the vertical guide roller causes the idler to pivot and force the belt back to its normal position. These idlers can be also adapted to flat carrying belts.

They should be intermittently spaced on both the carrying and return runs of the conveyor, to provide against injury to the edges of the belt due to misalignment. They are especially valuable for belts having unequal stretch or off loading and for belts subjected to side wind forces. They should be spaced at 25 to 50 feet from each end terminal or bend pulley and approximately 100 feet apart thereafter.

"Auto Tilt" Idlers.

These are used on reversible belt conveyors and shuttle belt conveyors. They consist of standard 3 or 5 pulley idlers, except that the end supports are equipped with rocker mountings which permits the idler set to tilt 2° in either direction. When the belt is driven forward the idler set tilts forward and reverses the tilt when running backward. Rubber bushed rocker mounting is usually adopted.

Impact Idlers.

On belt conveyors and feeders handling runs of mine coal, stone, ore and similar heavy materials, impact idlers are used to absorb or cushion the impact of heavy pieces of material as they fall on the belt at the loading or transfer point. The impact of such material tears the belt and pounds the idler sets which soon show signs of wear and damage. The chief purpose of the adoption of this type of idler is to prevent the carcass of the belt being ruptured, the rubber cover cut and damaged and to prolong the life of the idler set at this point of the conveyor.

The idler set shown in Fig. 10 is a design of The British Tyre & Rubber Company.

Each roll is made up of a steel tube in which tapered roller bearings, labyrinth seals and shafts are mounted. The tube has a collar securely fastened on one end to form a shoulder and the other end is threaded to take a special flanged nut. The rubber cushion is made of a series of individual interlocking rubber rings which, when assembled on the tube, form a series of alternately deep and shallow grooves to insure maximum impact absorption qualities. The surface of the rubber is provided with a diamond tread which, together with the grooves, retards side slip under

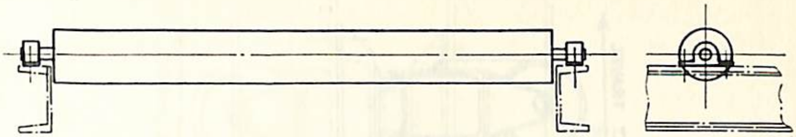
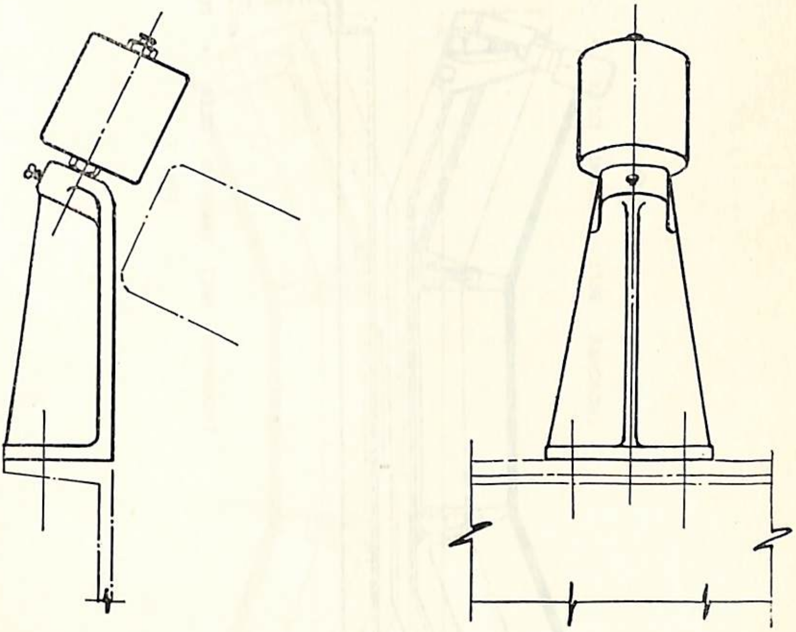


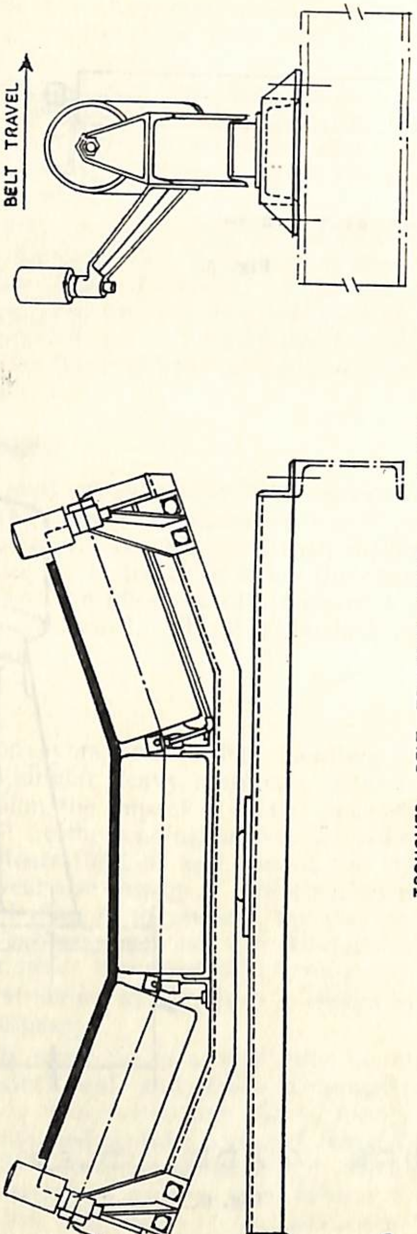
FIG 5 - FLAT BELT IDLER

Fig. 5.



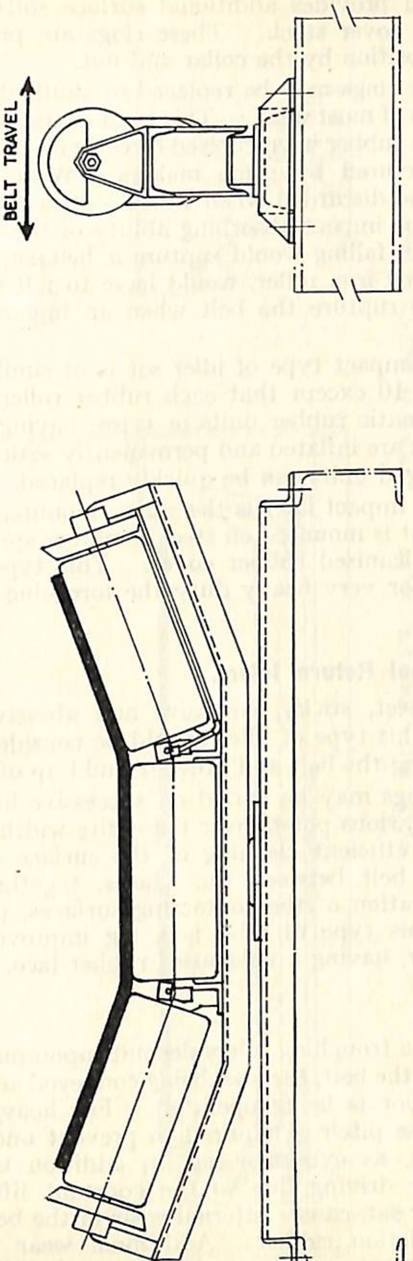
SIDE GUIDE IDLER

Fig. 6.



TROUGHED BELT TRAINING IDLER • POSITIVE ACTION TYPE

Fig. 7.



TROUGHED BELT TRAINING IDLER • COUNTERWEIGHTED - DISC TYPE

Fig. 8.

lateral blows, and provides additional surface softness for protection of the belt cover stock. These rings are pressed on the tube and held in position by the collar and nut.

When worn, the rings may be replaced or shuffled to place the least worn at points of most wear. This is, of course, an advantage over rolls where the rubber is vulcanised directly on the tube, which is a type manufactured by some makers. With solid rubber rollers they must be discarded when worn. It is claimed by the makers that the true impact absorbing ability of the idler, a piece of material which in falling would rupture a belt supported on a standard steel or cast-iron roller, would have to fall up to five or six times as far to rupture the belt when an impact absorption idler is used.

The pneumatic impact type of idler set is of similar design to that shown in Fig. 10 except that each rubber roller is made up of a series of pneumatic rubber units or tyres, having thick wear-resisting treads, and are inflated and permanently sealed to prevent loss of air. Damaged units can be quickly replaced.

Another form of impact idler is the spring mounted type. The whole of the idler set is mounted on steel or rubber springs and the rollers having a vulcanised rubber cover. This type is a good, cheaper form, but for very heavy duty the foregoing tyred types are recommended.

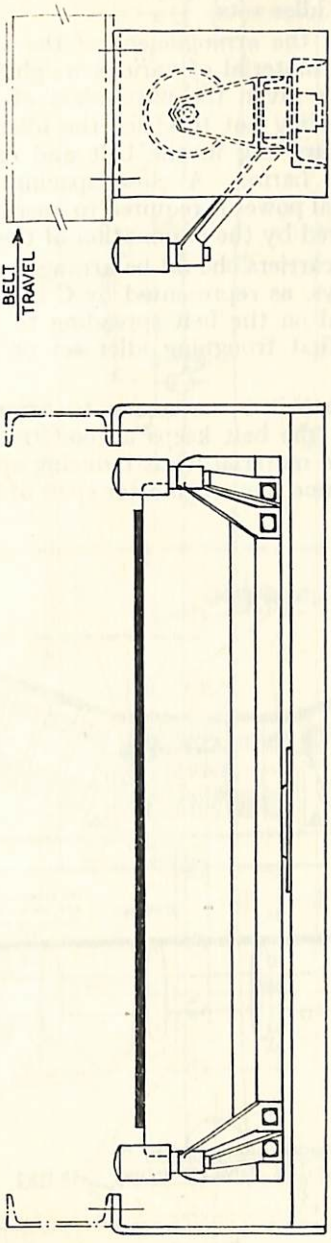
Rubber Covered Spool Return Idlers.

For use where wet, sticky, corrosive and abrasive materials are being handled, this type of idler should be considered. They greatly aid in cleaning the belt and prevent build up of material.

The wheel spacings may be varied on successive idlers so that the wheels contact various points over the entire width of the belt, providing for more efficient cleaning of the surface of the belt. The flexing of the belt between the wheels, together with the distortion or deformation of the contacting surfaces, provides the cleaning effect. This type of idler is a big improvement upon the plain return idler, having a vulcanised rubber face, see Fig. 10.

Spacing of Idlers.

The spacing of the troughing idlers depends upon many factors, such as the weight of the belt, the load being conveyed and the speed at which the conveyor is being operated. For heavy materials or high speeds a close pitch is required to prevent undue sagging between the carriers, as excessive sag, in addition to requiring additional power for driving due to the constant lifting of the material at each idler set, causes internal wear in the belt from the stretching of the friction rubber. Additional wear is also experienced on the rubber cover, due to the material shifting as it is



FLAT BELT TRAINING IDLER • POSITIVE ACTION TYPE

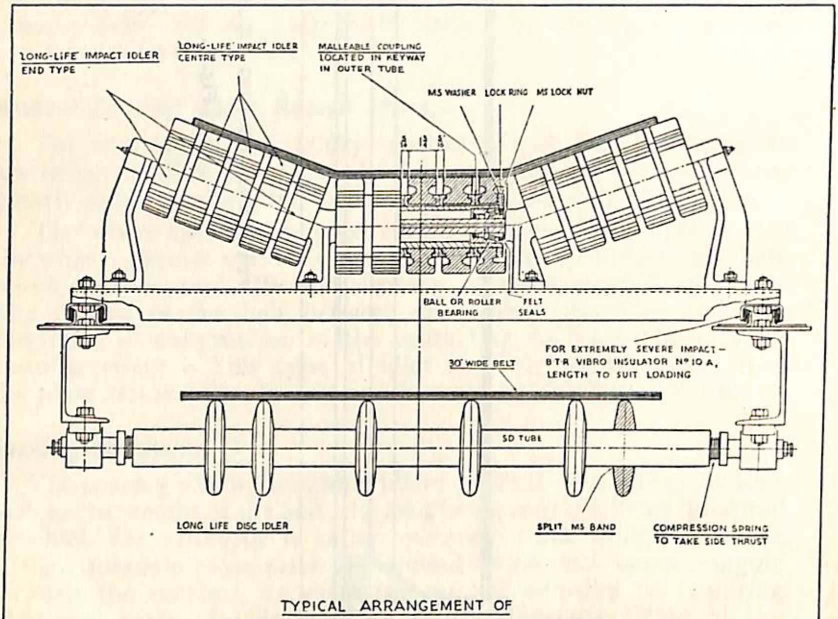
Fig. 9.

repeatedly flattened between the idlers and then troughed again when passing over the idler sets.

In Fig. 11 is shown the arrangement of the standard spacing of troughing rollers for material of various weights per cubic foot, and in Table No. 9 are given the particulars of the distances A and C. It is sound policy not to space the idlers too far apart, producing an unnecessary sag in the belt and causing a flexing that does considerable harm. A close spacing is much to be desired and no additional power is required to operate the additional pulleys, as power is saved by the elimination of the sag.

The end troughing carriers should be arranged close up to both the end terminal pulleys, as represented by C in Table 9, in order to prevent the material on the belt spreading to the edges either before it reaches the first troughing idler set or before it passes over the head pulley.

At the loading shoot it is necessary to arrange the carriers closer together so that the belt keeps a good troughed formation for the reception of the material, thus reducing spillage to a minimum. At the same time a much shorter span of belt is presented


'LONG-LIFE' IMPACT IDLER END TYPE
'LONG-LIFE' IMPACT IDLER CENTRE TYPE
MALLEABLE COUPLING LOCATED IN KEYWAY IN OUTER TUBE
MS WASHER
LOCK RING
MS LOCK NUT
BALL OR ROLLER BEARING
FELT SEALS
FOR EXTREMELY SEVERE IMPACT – B.T.R. VIBRO INSULATOR Nº 10 A, LENGTH TO SUIT LOADING
30" WIDE BELT
SD TUBE
LONG LIFE DISC IDLER
SPLIT MS BAND
COMPRESSION SPRING TO TAKE SIDE THRUST

TYPICAL ARRANGEMENT OF
B.T.R. LONG-LIFE IMPACT & RETURN IDLER DISCS

Fig. 10.

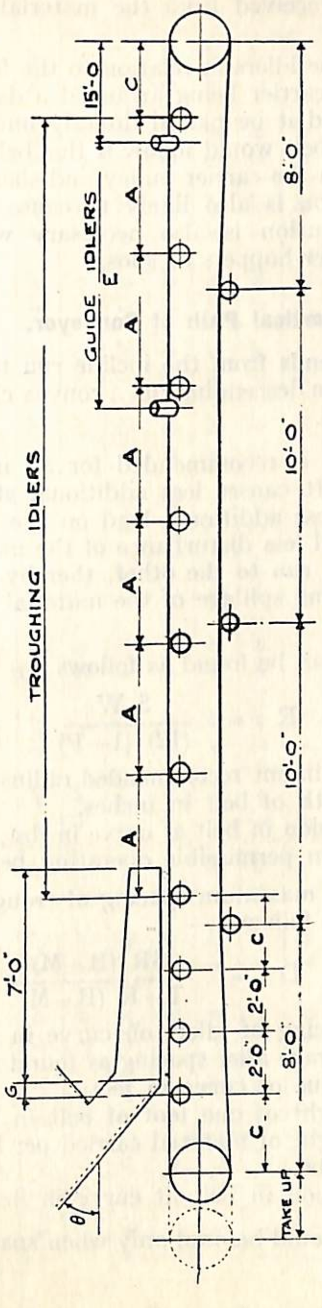

TROUGHING IDLERS
GUIDE IDLERS
15'-0
C
E
A
A
A
A
A
A
8'-0"
10'-0"
10'-0"
7'-0"
C
2'-0"
2'-0"
8'-0"
C
G
θ
TAKE UP

Fig. 11.

against the impact received from the material and deflection is thereby prevented.

The position of the idlers in relation to the feed point is shewn in Fig. 11, the first carrier being arranged a distance of 6″ back, and in no case should it be placed directly under the belt as increased wear on the belt would follow if the belt was continually being nipped between the carrier pulley and sharp heavy lumps of material. This action is also likely to cause the carrier to be broken. This precaution is also necessary where a travelling filler is arranged under hoppers or silos.

Convex Changes in Vertical Path of Conveyor.

Where the belt bends from the incline run to a horizontal run or a horizontal run to a descending run a convex change in the travel of the material occurs.

The convex curve is recommended for all installations where space will permit. It causes less additional stress in the edges of a troughed belt, less additional load on the end rollers of the idlers at the bend and less disturbance of the material on the belt as it passes from one run to the other, thereby reducing wear on the belt and preventing spillage of the material over the edges of the belt.

Minimum radius can be found as follows :—

$$R = \frac{5\ W}{(12)\ (1-P)}$$

Where R = minimum recommended radius of curve in feet.
W = width of belt in inches.
P = tension in belt at curve in lbs., divided by maximum permissible operating belt tension in lbs.

The recommended maximum spacing of troughed idlers on the curve can be found as follows :—

$$D = \frac{SR\ (B+M)}{T+R\ (B+M)}$$

Where D = Spacing of idlers on curve in feet.
S = average idler spacing as found from Table 9.
R = radius of curve in feet.
B = weight of one foot of belt in lbs.
M = weight of material carried per linear foot of belt in lbs.
T = tension in belt at curve in lbs.

The Bend Pulley should be used only when space will not permit a convex curve.

TABLE No. 9.

STANDARD SPACING OF IDLER SETS.

Weight of Material lbs. cub. ft.		Width of Belt in Inches.																	
		12	14	16	18	20	22	24	26	28	30	32	34	36	40	44	48	54	60
50	A	5′-0″	5′-0″	5′-0″	4′-6″	4′-6″	4′-6″	4′-6″	4′-6″	4′-6″	4′-0″	4′-0″	4′-0″	4′-0″	4′-0″	3′-9″	3′-9″	3′-6″	3′-6″
	C	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	2′-6″	2′-6″
75	A	5′-0″	5′-0″	4′-6″	4′-6″	4′-6″	4′-6″	4′-6″	4′-6″	4′-6″	4′-0″	4′-0″	4′-0″	4′-0″	4′-0″	3′-9″	3′-9″	3′-6″	3′-3″
	C	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	2′-9″	2′-9″	2′-9″	2′-9″	2′-9″	2′-6″	2′-6″	2′-6″	2′-6″
100	A	4′-6″	4′-6″	4′-6″	4′-6″	4′-6″	4′-6″	4′-0″	4′-0″	3′-6″	3′-6″	3′-6″	3′-6″	3′-6″	3′-6″	3′-3″	3′-3″	3′-0″	3′-0″
	C	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	3′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″
125	A	4′-6″	4′-6″	4′-6″	4′-0″	4′-0″	4′-0″	4′-0″	3′-6″	3′-6″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″
	C	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″	2′-6″
	E	30′-0″	30′-0″	30′-0″	35′-0″	35′-0″	40′-0″	40′-0″	40′-0″	40′-0″	40′-0″	45′-0″	45′-0″	45′-0″	45′-0″	45′-0″	45′-0″	45′-0″	50′-0″
Picking	A	—	—	—	—	—	—	3′-6″	3′-6″	3′-6″	3′-6″	3′-6″	3′-6″	3′-0″	3′-0″	2′-6″	2′-6″	2′-6″	2′-6″
Belts	C	—	—	—	—	—	—	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	3′-0″	2′-6″	2′-6″	2′-6″	2′-6″

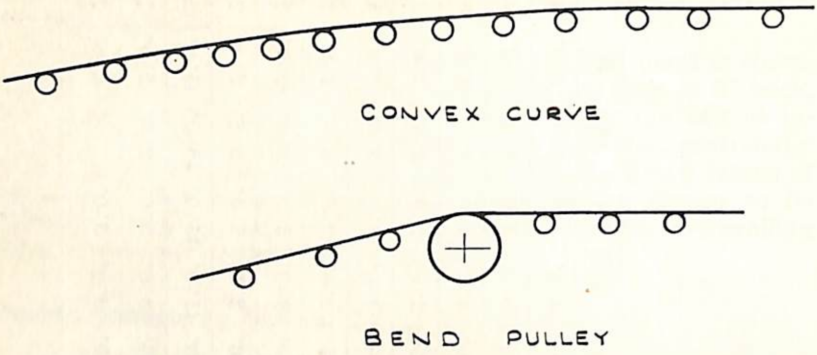


Fig. 12.

The diameter of the bend pulley should be large enough to insure the material maintaining contact with the belt as the belt changes direction.

The diameter required varies with the cosine of the angle of change in direction and the square of the speed of the belt, and becomes quite large for belt speeds over 500 feet per minute, as shown in Table No. 10.

The spacing of the idlers should be half standard spacing for troughing idlers as found for the belt.

TABLE No. 10.

Dia. Bend Pulley inches.	Speed in ft/min.	Dia. Bend Pulley inches.	Speed in ft/min.
16	200	36	400
20	300	54	500

Return Idlers.

The usual spacing for return idlers is from 8′-0″ to 10′-0″ apart, as shewn in Fig. 11, depending upon the weight of the belt, and as close as possible after all cross supports in order to protect the return belt from injury. Sufficient clearance should also be allowed between the floor and the underside of the pulleys so that any material spilled from the conveyor will not collect and foul the pulleys.

Return idler pulleys should in no case be arranged close enough to the drive pulley to act as a snub pulley. If increased arc of contact is required on the driving pulley, or if it is necessary to raise the belt over a cross support close to the drive end, or where the belt bends from the incline to the horizontal, use a snub pulley.

Package Conveyor Idlers.

The spacing of the idlers for package conveyors varies according to the size and weight of the package and also the width of the belt. For ordinary conditions the figures in Table No. 11 may be taken as good general practice for medium sized packages of average weight. For light parcels the pitch may be slightly increased, but for heavy boxes, crates and assembled machine parts the centres of the rollers should be spaced at least half the length of the article so that the load is always being supported by two rollers, thus eliminating undue stretching and deflection of the belt. For large heavy cumbersome articles the belt conveyor may prove costly and the loading and unloading assume a question of great importance. A small overhead runway would probably better meet the conditions, or a slat conveyor.

TABLE No. 11.

SPACING OF IDLERS FOR PACKAGE CONVEYORS.

Width of belt in inches.	Carrier Idlers.		Return Idlers. 2″ and 4″ diam. Idler Rollers.
	Approx. 2″ diam. Idler Rollers.	Approx. 4″ diam. Idler Rollers.	
14″	4′-6″	—	12′-0″
16″	4′-6″	—	12′-0″
18″	4′-0″	4′-0″	10′-0″
20″	4′-0″	4′-0″	12′-0″
24″	3′-6″	3′-6″	10′-0″
30″	3′-6″	3′-6″	10′-0″
36″	—	3′-0″	10′-0″
42″	—	3′-0″	8′-0″
48″	—	3′-0″	8′-0″

TRAINING THE BELT.

In the early days of belt conveyors difficulty was sometimes experienced in keeping the belt in alignment. This was chiefly due to the deep troughing effect by the 45° carriers and with the adoption of shallower troughing much of this trouble was eliminated. Correct alignment of the carriers is one of the most important factors in order to effect a true running belt.

The supporting stands of the carrier pulleys are generally provided with slot holes in the base so that each individual set can be adjusted to the run of the belt. To line up the carriers, one side should be only partly tightened down in position, and by running the conveyor empty any tendency of the belt to run either to left or right can be corrected by tapping the carriers forward or backward until the belt is completely adjusted to a central running position.

Always start work in the direction in which the belt travels. The pulley which is causing climbing of the belt may be the second or third set from the point where climbing occurs and this point should be carefully noted.

It is essential that correct alignment be obtained, as belts which are allowed to ride over the edges of the outer carrier pulleys against the sides of the shoots or other structures in close proximity will rapidly be worn away at the edges.

The side guide rollers are too often relied upon to keep the belt straight, but usually they do more harm than good.

With the old form of deep troughing they were essential, but as the belt reduces in troughing width so does the need for guide rollers diminish, until for flat belts their adoption is not at all necessary, providing correct alignment of the carriers is obtained.

Narrow belts are more troublesome than wide belts regarding true running and it is customary to provide side guide rollers as a help to maintain proper alignment in long conveyors.

Wide belts do not require side guide rollers, unless the operating conditions are adverse to true running.

The effect of the side guide roller is not so marked with a wide belt as in the case of a narrow belt, due to the small proportion of width acted upon, and as previously stated bending or folding is more apt to occur than actual guiding.

To meet the contingency of keeping the belt in correct alignment various devices have been adopted. In Fig. 13 is shewn an early method employing a troughed carrier set with the outer pulleys arranged with a rake forward in the direction of the travel of the belt. The main feature of this device is that as the belt travels over the pulleys it has a tendency to converge to a distant central point, proportionate to the angle at which the outer rollers are set. Although this works quite well, it has not found great favour and is not now very largely used.

Another method employed in Fig. 13, is to bevel the cross bearer on the top face so that the carrier set is tilted forward in the direction of travel of the belt, thus producing a somewhat similar effect as the special idler set, shewn in Figs. 7 and 8. Should the belt ride to one side or the other, the side rollers tend to force it back again to the centre. This is a very simple device, causing very slight additional wear on the underside of the belt, and is therefore finding great favour as a measure for keeping the belt in alignment.

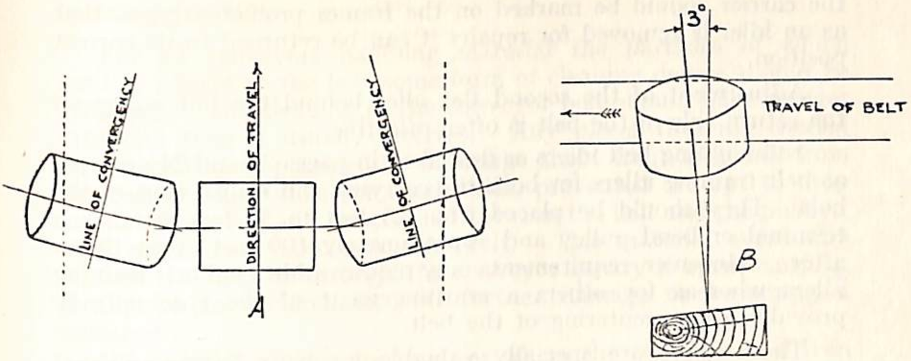


Fig. 13.

A method which is very often used by the uninitiated and which causes considerable injury to the belt is that of applying additional tension on the take-ups or weighted tension. This may seem an easy way out of the difficulty, but the additional tension imposed on the belt will cause excessive stretch and may result in the splice breaking apart.

There are many causes which result in the belt not running true, such as :—

(1) Feeding the material sideways on to the belt instead of in the direction of travel, thus causing uneven loading and consequent shifting of the material as it passes over the carrier sets.

(2) Idler pulleys becoming tight or sluggish on one side, thus causing uneven deflecting forces.

(3) The carrier sets not being in alignment.

(4) The splice not being arranged at right angles.

(5) Too much sag between the carrier sets, due to the pitch of the carriers being too large for the weight of the material conveyed.

It is essential for the smooth running of the conveyor that all the points mentioned receive careful consideration.

Misalignment.

If a belt keeps climbing sideways on the same idlers, the fault is not with the belt, but with the alignment of the idlers. In lining up idlers, work in the direction of belt travel. In adjusting idlers to align the belt, they should always be kept at right angles with the centre line of the belt. One operator finds that it pays to lay out the centre line with a transit. The correct location of the carrier should be marked on the frames prominently, so that as an idler is removed for repairs it can be returned to its correct position.

Adjustment of the second flat idler behind the tail pulley on the return side of the belt is often effective.

Self-aligning belt idlers as described in pages 22 and 24 are used as belt training idlers for both the carrying and return runs of the belt. They should be placed from 25 feet to 50 feet from each terminal or bend pulley and approximately 100 feet apart thereafter. However, requirements may require additional belt training idlers whereas for others a rearrangement of the spacing may provide proper centering of the belt.

These idlers are specially valuable for belts having unequal stretch or off centre loading and for belts subjected to side wind forces.

Protecting the Belt.

As already pointed out, every precaution and protection should be given to the belt to prevent any damage occurring. This is absolutely essential if long life is to be obtained and the fact that the cost of renewing the belt is in the majority of cases a heavy item, confirms the necessity of giving close attention to every action which is liable to cause destruction or wear to the belt.

Dirt Tight Decking.

To prevent any material which may fall off the carrying run of the belt from dropping on to the return run of belt dirt tight decking is invariable used. This decking, which is usually constructed of light gauge steel sheet, is arranged between the two belt runs fixed to the conveyor stringers.

Housing.

To protect the conveyor belt from the elements it is advisable to provide some form of cover or housing. They are also used to enclose the belt conveyor and provide accessibility where the conveyor is carried on a bridge or gantry as when it is required to span open spaces. Many forms of galleries and housings are possible depending upon the conditions encountered.

Fig. 65 shows a horizontal or inclined open gantry with walkway arranged on one or both sides and cover provided over the conveyor only. Side sheeting if provided would act as wind boards to prevent the belt being lifted and damaged, which very often happens when exposed to high winds. Figs. 63 and 64 show closed gantries with the walkways on one or both sides of the conveyor.

Cleaning the Belt.

For all conveyors handling material the particles of which tend to adhere to the belt some form of cleaning device should be arranged. The cleaning of the conveyor belt should have greater attention than it usually receives. Any pieces that are carried on the belt after passing the discharge shoot frequently fall from the belt on the return run, and unless prevented are apt to drop on the return belt and get crushed between the belt and the pulleys, with subsequent damage to the belt. This material eventually falls on the floor or through the conveyor gantry if open, causing, if allowed to accumulate, still further damage or becoming a nuisance.

Where snub pulleys are used, or where the conveyor is driven by means of a tandem drive and the working face of the belt comes into contact with the pulley, it is essential that some means of cleaning the belt be provided to prevent the material accumulating and becoming encrusted on the pulley rim, causing damage to both the pulleys and the belt. Misalignment of the belt can also be traced to this cause.

Brushes.

Stationary brushes have been tried for cleaning belts, but without success, due to the bristles soon becoming choked, rendering the device useless. The latest and most general practice is to adopt revolving brushes, rotating in an opposite direction to that of the belt. The brushes, which consist of rattan or bassine fibre glued into wooden cylinders, should be made to stand up to hard service and at the same time cause no injury to the belt.

The speed of the brush should be sufficient to throw the fine material out of the bristles, as at slow speeds it has been found that the bristles soon make up. Some makers provide brushes with bristles arranged in spaced rows horizontally on the wooden cylinder and others arrange the bristles to form a continuous surface.

A cleaning brush with the bristles arranged in rows placed spirally round the wooden centre has been found to function very satisfactorily and is more self-cleaning than the other forms mentioned.

The usual diameter of cleaning brushes over the tops of the bristles varies from 8″ to 12″. It is not recommended to use brushes less than 8″ in diameter, owing to the small circumferential surface and the trouble of such small brushes making up with material.

In Table No. 12 is given the usual speeds for brushes to suit various materials.

TABLE No. 12.

Class of Material.	Peripheral Speed of Brush F.P.M.			R.P.M. of Brush.		
	8″	10″	12″	8″	10″	12″
Dry, Dusty.	800	870	1000	380	330	320
Damp.	960	1100	1200	455	420	380
Wet, Sticky.	1200	1380	1500	—	525	475

Fig. 14 illustrates a standard revolving brush cleaner suitable for dry materials operating at the slower speeds, with a dust-tight spur gear driving mechanism on a pivoted mounting.

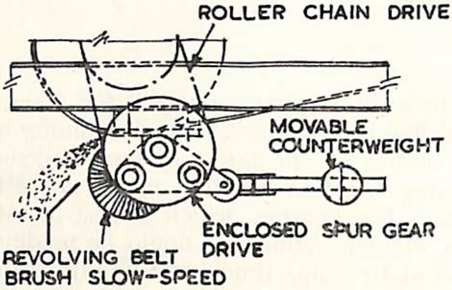


Fig. 14.

Fig. 15 shows the high speed revolving brush which is recommended where the material has a tendency to stick to the brush. The surface, rotated at a speed of approximately 1200 feet per minute, keeps itself clean by centrifugal action. An adjustable counterweight used on both types maintains proper brush pressure and compensates for wear on the bristles.

Special beater brushes are sometimes used to dislodge caked or frozen material from belt. They consist of reinforced rubber strips which take the place of the bristles. Peripheral speed is approximately 700 F.P.M.

The least expensive and simplest type of belt cleaner is the balanced blade type cleaner. This cleaner usually consists of a rubber strip suitably supported and fitted with balance weights on lever arms.

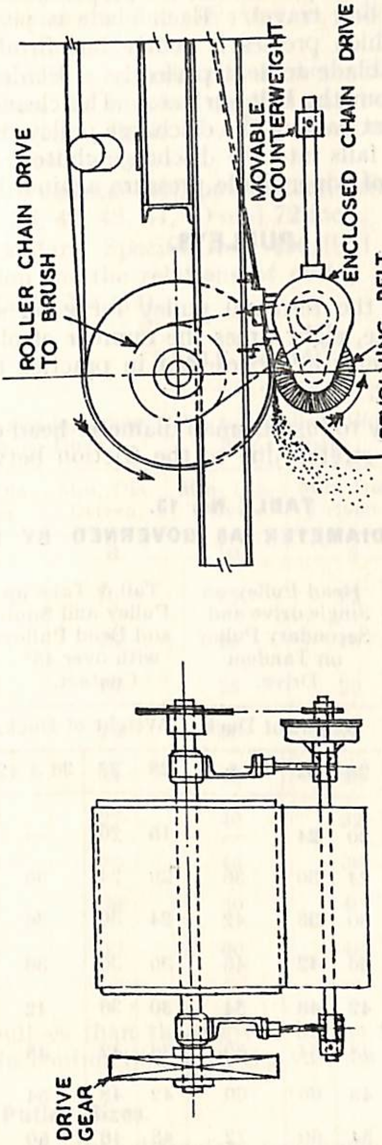

ROLLER CHAIN DRIVE TO BRUSH
MOVABLE COUNTERWEIGHT
ENCLOSED CHAIN DRIVE
REVOLVING BELT BRUSH HIGH-SPEED
DRIVE GEAR

Fig. 15.

The Stephens Adamson Mfg. Co., Ltd., have a multiple blade cleaner used for removing wet or dry material. It consists of a row of thin spring steel wiper blades, set perpendicular to the belt, but diagonal to the belt travel. Each blade is pivoted upon an individual spring, which presses it gently but firmly against the belt and allows the blade to seat perfectly. Blades overlap and wipe the material from the belt surface. The cleaner is mounted on the return run, just back of the discharge pulley, where material wiped from the belt falls into the discharge chute. A hand lever permits adjustment of wiper blade pressure against belt.

PULLEYS.

Head Pulleys.

The diameter of the terminal pulley for good service should be as large as possible, eight times the number of plies of the belt being the ideal diameter in inches, but in practice these sizes are very seldom adopted.

It is false economy to install small diameter head or tail pulleys, as there is excessive stretch due to the friction between the plies

TABLE No. 13.

NORMAL PULLEY DIAMETER AS GOVERNED BY BELT PLIES.

No. of Plies.	Primary Pulley on Tandem drive and Head Pulley for Tandem Driven Conveyor			Head Pulley on Single drive and Secondary Pulley on Tandem Drive.			Tail & Take up Pulley and Snub and Bend Pulleys with over 45° Contact.			Snub and Bend Pulleys with less than 45° Contact.		
	Weight of Duck.			Weight of Duck.			Weight of Duck.			Weight of Duck.		
	28	32	36 & 42	28	32	36 & 42	28	32	36 & 42	28	32	36 & 42
4	24	30	—	20	24	—	18	20	—	12	15	—
4	30	36	42	24	30	36	20	24	30	15	18	24
6	36	42	48	30	36	42	24	30	36	18	24	24
7	42	48	54	36	42	48	30	36	36	24	24	30
8	48	54	60	42	48	54	30	36	42	24	30	36
9	54	60	72	48	54	60	36	42	48	30	36	36
10	60	72	72	48	60	60	42	48	54	30	36	42
11	60	72	84	54	60	72	48	48	60	36	42	48
12	72	84	84	60	72	72	48	54	60	36	42	48

of the belt, and with short conveyors this repeated bending causes the rubber to suffer from fatigue and the plies to come apart. With small pulleys a greater tension is also necessary, although the ratio of tensions remains the same, in order to obtain sufficient adhesion to the pulley face.

General practice is to make the diameter in inches of all driving pulleys five times the number of plies and tail pulleys four times the number of plies. Table No. 13 gives normal pulley diameters as governed by belt plies.

It is usual to use standard pulley diameters wherever possible—12, 18, 24, 30, 36, 42, 48, 54, 60 and 72 inch.

British Standard Specification 490/1951 gives the following recommendation for the relations of pulley diameter to belt construction. (Table No. 14).

TABLE No. 14.

No. of Plies.	Up to and including 32 oz. Duck.		Up to and including 36 oz. Duck.		Up to and including 42 oz. Duck.	
	Min. Dia. Driver.	Min. Dia. Driven.	Min. Dia. Driver.	Min. Dia. Driven.	Min. Dia. Driver.	Min. Dia. Driven.
2	10	6	10	8	12	10
3	15	10	15	12	18	15
4	18	15	20	16	24	20
5	24	18	25	20	30	25
6	27	21	30	24	36	30
7	30	24	35	26	42	35
8	36	27	40	32	48	40
9	42	30	45	36	54	45
10	48	36	50	40	60	50
12	54	42	60	48	72	60

Smaller pulleys than those given in the foregoing table should not be used in conjunction with the various plies of belt specified.

Selection of Pulley Sizes.

Table No. 13 gives diameters recommended for belt conveyor drive, tail and snub pulleys. Pulleys of too small a diameter may cause a separation of plies or even break the fabric in outer plies due to the excessive stress caused by stretching the outer plies in making sharp bends.

Rubber lagged pulleys increase the tractive effort of the plain driving pulley on a belt from 10% to 20% where contact between the pulley and the belt is clean, or where the dust from the material handled is slightly damp. In dry and very dusty conditions, when handling coal, clay and similar smooth materials, the adhesion of the fine particles to the lagging usually decreases the tractive effort by 10% to 20%. It is therefore advisable to study the condition under which the conveyor is required to operate before resorting to the use of lagged pulleys.

Flanged pulleys as a means of making the belt run centrally should not, under any consideration, be used. A heavy crown on the pulley face of at least $\frac{1}{8}''$ per foot width will keep the belt well centered, as it will always ride to the top of the crown. Some makers prefer an even more pronounced crown than $\frac{1}{8}''$, but in ordinary cases where the conditions are quite normal, this should prove quite sufficient to centre the belt.

The head pulley should always be of heavy construction to withstand the high stresses imposed at slow speeds, perfectly true, and for belts over 24″ wide should have double arms. The width of the face should in no case be less than 2″ wider than the belt and the following are the proportions generally attained in good practice :—

Belts up to 48″	... ...	Pulleys 2″ wider.
Over 48″	... ...	,, 3″ ,,

This extra width permits the belt to run to either one side or the other, particularly when first starting up, and during ordinary working if a slight disalignment should occur, and will fully recompense the small extra cost by protecting the belt from injury to the edges and will in many cases relieve the system of side guide rollers.

The diameters given in Table 13 are the minimum that should be used where the belt is to be stressed to the full tension rating of the duck. If the belt is not operated under full allowable tension, pulley diameters may be reduced in proportion to the reduction in tension. It is never advisable to reduce diameters to less than 75% of the normal sizes given.

The pulley sizes listed are intended for normal design with the drive and maximum tension at the head end. Where the maximum tension is greatest at tail end, as in retarding conveyors, tail pulleys should be larger.

The tail pulley is usually of similar construction to the head or driving pulley, being cast-iron or wrought-iron as circumstances warrant. The same crowning of the face should be provided, but the diameter is usually smaller, generally four times the number of plies of the belt.

Where the conditions as previously pointed out do not suit the adoption of rubber lagging on the driving pulley, a snub pulley is used to increase the arc of contact, thus increasing the driving grip of the belt. In the ordinary case, where no snub pulley is employed, the arc of contact is approximately 180°, but with the adoption of a snub pulley, this is increased to 240°, as shewn in Fig. 39.

As shewn under "Tension in Belts," the ratio of tension for an ordinary bare iron pulley with a belt wrap of 180° = 2·19, and for the same case at 240° the ratio = 2·85. Taking the effective driving tension $T_1 - T_2$ at 1000 lbs., the tension in the belt for the first case equals

$$\frac{T_1 - T_2}{T_2} = \frac{2{\cdot}19 - 1}{1}$$

Therefore $T_2 = 1000/1{\cdot}19 = 840$ lbs.,
and $T_1 = 1000 + 840 = 1840$ lbs.

In the second case $T_2 = 1000/1{\cdot}85 = 540$ lbs.
and $T_1 = 1000 + 540 = 1540$ lbs.

The resultant decrease of the tension in the belt equals 300 lbs., or approximately 20% which would enable a 4-ply belt to pull as much as a 5-ply belt, with belts of other plies in proportion.

This is the great advantage of installing a snub pulley, but care should be exercised in this connection to always maintain a sufficient number of plies for proper flexibility and good service. The extra tractive pull which can be obtained by the addition of a snub pulley may be used to extend the length of a fully-loaded conveyor approximately 30%, providing of course the shafts and gears are increased in strength proportionately. The disadvantages of the use of a snub pulley are the reverse bending of the belt and the fact that any material adhering to the top cover is pressed into the rubber cover when passing over the snub pulley. In the latter case, some form of cleaning device should be provided to eliminate this.

In ordinary cases a snub pulley is usually not necessary, but for belts handling heavy materials or doing heavy duty the snub pulley is generally required. Where the necessary driving tension can be obtained without installing such a pulley it is advisable to eliminate same and at the same time save expense.

The width of the snub pulleys should not be less than 2″ wider than the width of the belt.

When using deflector pulleys on foot-driven conveyors, the pulleys around which the driving or tight side of the belt passes should not be less in diameter than the end driving pulley, that is, 5″ diameter per ply of belt. The other pulleys engaging with the slack side can be made the same diameter as given for snub pulleys, but preferably 3″ to 4″ diameter per ply of belt. It is a good rule with all pulleys to make them as large as possible.

Pneumatic Pressure Pulleys.

Pneumatic pressure pulley is a pneumatic snub pulley of Stephens Adamson Manufacturing Co., U.S.A., which increases traction between the drive pulley and the conveyor belt, and often eliminates the need for a tandem pulley drive.

The usual practice in driving a conveyor belt is to apply the power through the head pulley. This requires initial tension in the belt to give the drive pulley the necessary traction. With long heavily loaded belts, the initial tension to obtain traction often requires several extra plies of duck.

With the pressure pulley and single drive pulley only enough initial tension is required to prevent the belt sagging between the carriers.

The close position of the pressure pulley also gives the conveyor belt a greater arc of contact with the drive pulley—an increase of 15% to 20% over that given by the conventional snub pulley. The resilient pneumatic action also permits the use of any standard type of belt fastening.

The advantages claimed are as follows :—

Increased traction between drive pulley and conveyor belt.
Reduced initial belt tension.
Lighter and less expensive conveyor belt.
Longer belt life due to less flexing.
Elimination of tandem drive in many cases.
Reduced original conveyor costs.
Adjustable pulley pressure to suit conditions.
Belt tension factors K under section slack tension can be reduced 50% when pressure pulley is used.

The construction of the pulley consists of a series of standard pneumatic units similar to car tyres, as described for pneumatic impact idlers. Equalized air pressure is applied to all units by means of a valve on one side. As shown in illustration No. 16, the pressure against the drive pulley can be regulated, decreased for normal conditions or increased when the belt tends to slip due to dampness, atmospheric conditions or heavier loads.

The diameter A varies from 16½ inches for 18″ to 22″ wide belts, to 22″ dia. for belts 24″ wide and over.

Magnetic Separator Pulleys are used for removing tramp iron rom material that can be carried upon belt conveyors.

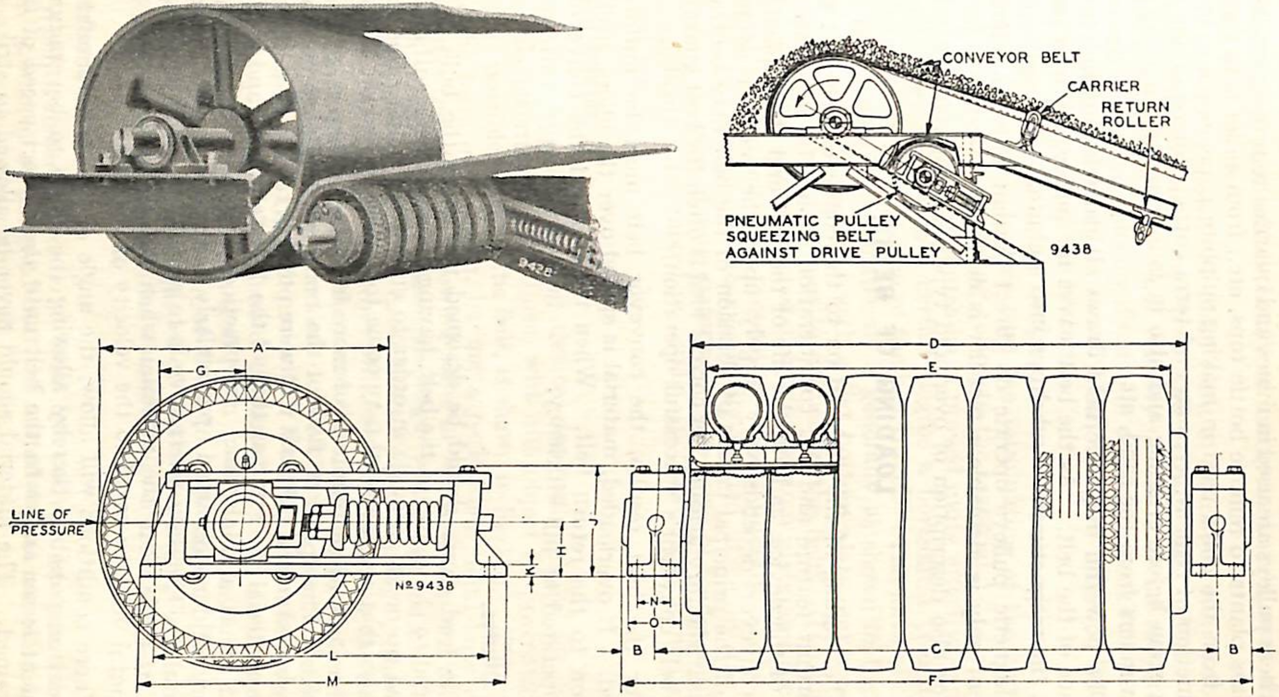

9428
CONVEYOR BELT
CARRIER
RETURN ROLLER
PNEUMATIC PULLEY SQUEEZING BELT AGAINST DRIVE PULLEY
9438
A
G
LINE OF PRESSURE
Nº 9438
J
H
K
L
M
D
E
N
O
B
C
B
F

Fig. 16.

These pulleys are used in mines and quarries to protect crushers; in glass plants to remove bottle tops, etc., from cullet; to remove iron from the chips used in making sulphite paper stock; in the production of cement, gypsum and terra cotta; for removing iron from grain before grinding and also in foundries for removing gits and runners from the sand, etc.

The powerful magnetic field draws all particles of iron to the surface of the belt. As the belt leaves the pulley, the iron passes from the magnetic field and drops into a chute.

Magnetic pulleys operate on direct current only and if A.C. current only is available a rectifier is necessary.

See also description of overband type separators.

LOADING OF BELTS.

The loading of the material on to the conveyor belt is a very important feature and full consideration should be given to this part of the device, as both the life of the belt and the capacity of the conveyor depends very largely upon the method adopted. One of the principal causes of abrasion of the belt may be traced to unsatisfactory loading. A correct feed is often the best guarantee of a belt conveyor's successful operation.

For the best results, the conveyor belt must be uniformly loaded. If overloaded, material is spilled over the idlers, decking and on to the return belt. When underloaded the conveyor is not operated at full efficiency.

Load Chutes.

The feed chute should be designed to deliver the bulk of the material to the centre of the belt, having wings or leads sufficiently opened up or at an angle in order to enable the belt to free itself and also that the material may come to a state of rest on the belt.

The slope of the chute is of more importance than is generally realised. The speed or flow of the material should approximate to the speed of the belt and if it were possible to impart a velocity to the material the same as that of the belt, abrasion and cutting would be at a minimum. In practice, however, this condition is very seldom obtained, particularly with wet sand or sticky materials. The only alternative is to provide the feed chute with such an angle that the material when flowing down acquires a horizontal velocity equal to the velocity of the belt.

Where conditions will allow, the angle of the chute should be as great as possible, thereby allowing the chute to be gradually placed at lesser angles to the belt until the correct speed of flow is obtained. The material should never be allowed to strike the belt vertically, particularly heavy or lumpy materials, such as run of mine coal or ores.

In Fig. 17 is shewn the plain form of chute generally adopted for screened and sized materials and cereals, etc. Where long chutes are necessary the final velocity of the material may result in excessive vertical velocity if made in one continuous run and the chute shewn in Fig. 17 is usually employed to effect a braking action on the speed of the material.

Another form of chute designed with the same object is that shewn in Fig. 18. In this case the termination of the chute is curved to impart a retarding action to the material.

Both these forms of chutes have proved quite effective, and in practice they impart to a certain degree a horizontal velocity to the material in the direction of the belt travel. The form of chute used for loading heavy materials is generally as shewn in Fig. 19. Here the chute is provided with a flat bottom plate, cut away in the centre to allow the fines to deliver to the belt. The lumps, if of very large size, are arrested by the bottom plate and the fine material on the belt thus preventing impact directly on the belt, and the heavier pieces are eventually carried along by the belt resting on a layer of fine material. This type of loader finds its chief application for mixed materials, such as run or mine coal or unsized ore.

Screen Chutes.

Another type of loading device designed chiefly for heavy materials intermixed with fines, with the object of preventing as much as possible wear on the belt, is shewn in Fig. 20. This chute is in the form of a screen, having a perforated metal base plate, which has the effect of allowing some of the fine material to drop through on to the belt before the larger pieces are fed on. It is the large-sized pieces which cause most wear or damage ; therefore if the fines are loaded first, they will form a cushion for the impact of the large lumps and at the same time form a foundation upon which the lumps become bedded. With such a device the larger pieces do not touch the belt and much abrasion is eliminated.

Another point which should be noted is that the lip of the feed chute should be arranged with sufficient clearance above the belt to allow the layer of fines to pass under. Failure to observe this will result in pieces of material wedging between the belt and the chute and serious damage will occur to the belt.

Transfer Chutes.

When transferring material from one conveyor to another, the chute should be so designed that the material does not fall directly on to the second conveyor belt.

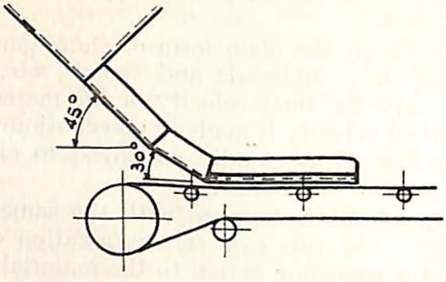


Fig. 17.

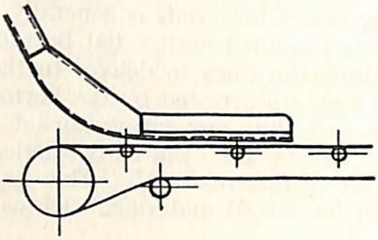


Fig. 18.

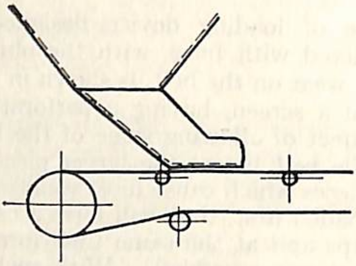
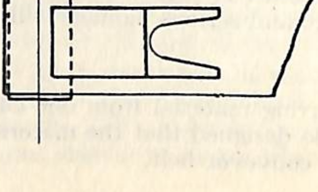


Fig. 19.



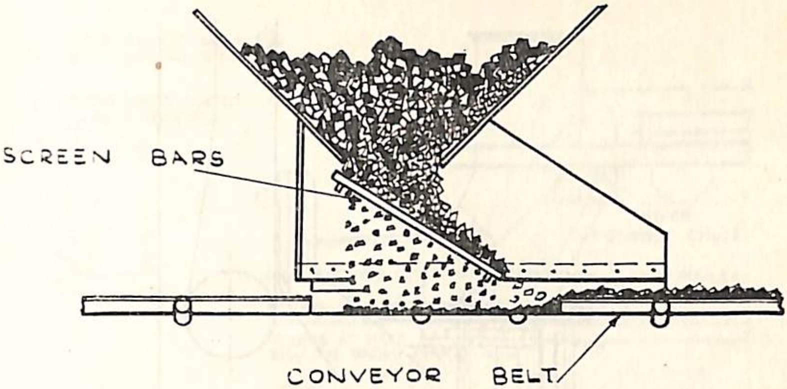


Fig. 20.

In Fig. 21, is shewn a typical design of an inclined conveyor arranged to feed to either or both of two horizontal conveyors. The feed or transfer chute is fitted with a swing plate valve, which can be thrown over to left or right at will to feed either conveyor, or moved to a central position if loading of both conveyors simultaneously is desired. The material, as it descends from the feeding conveyor, falls on the inclined baseplate of the chute, being thereby guided on to the horizontal belt in the direction of travel of same.

Side feeding to a belt, which is sometimes resorted to to save height, is a bad policy. Considerable damage, such as abrasion and cutting of the belt, is sure to follow and even if long skirt boards are adopted, which in themselves are deplorable, spillage is apt to occur. Another objection to side feeding is the misalignment of the belt due to its uneven loading. When passing over the carrying idlers the material tends to even itself on the belt, causing additional friction to the belt cover.

The object to strive for is to place the material on the belt with as little boiling action as possible before the material comes to rest, as this has a serious effect on the belt and is the cause of a good deal of spillage.

Fig. 22 shows a transfer chute from one belt conveyor to another in direct line. Here is shown a dribble chute to feed the material removed by the balanced scraper on to the preceding conveyor. When handling wet or damp material it is essential to use some form of belt cleaner and it is also necessary that the material is collected and diverted to the receiving belt or chute, otherwise in a very short time the material will accumulate into a large pile and cause untold trouble.

The angle of the chute should be as steep as possible to allow the material to flow off and thus prevent it holding up. Height

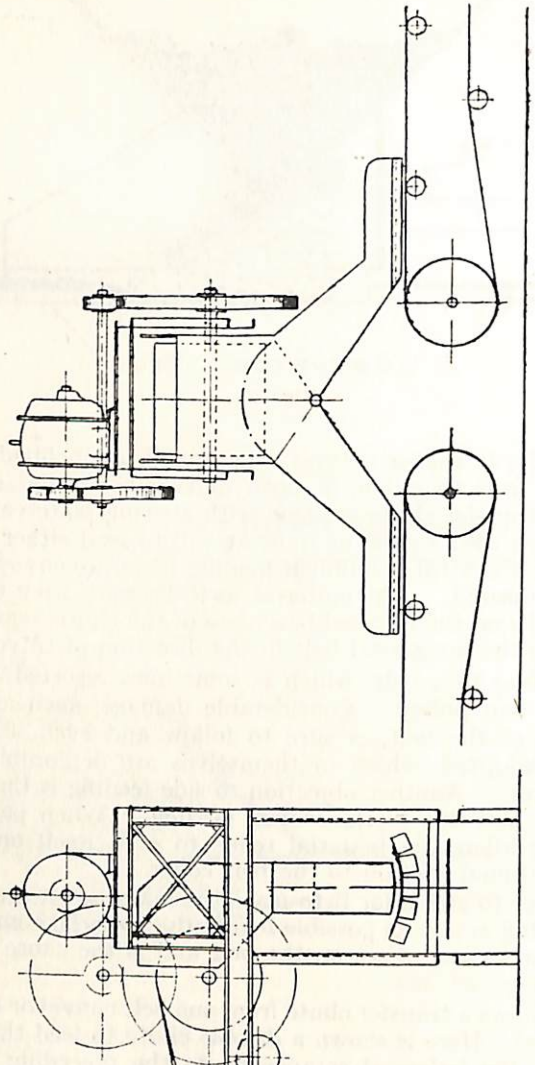
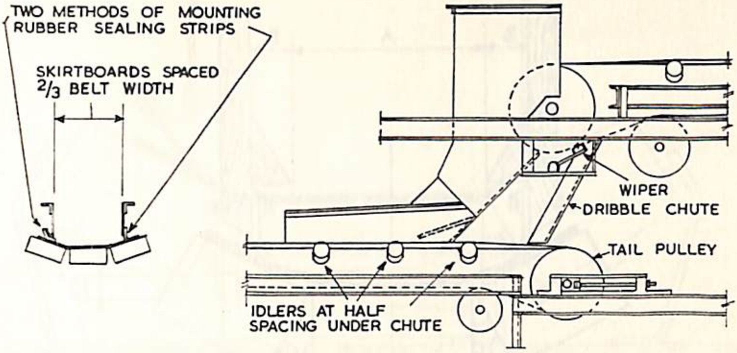


Fig. 21.


TWO METHODS OF MOUNTING RUBBER SEALING STRIPS
SKIRTBOARDS SPACED 2/3 BELT WIDTH
WIPER
DRIBBLE CHUTE
TAIL PULLEY
IDLERS AT HALF SPACING UNDER CHUTE

TRANSFER CHUTE WITH DRIBBLE CHUTE

Fig. 22.

is needed to design these chutes efficiently and a saving of height in the design stage is apt to become a serious nuisance when the conveyors are installed and put to work.

With rotary brushes the material can be fed into the direct stream of material in the transfer or discharge chute.

Skirt Boards.

The object of skirt boards is to confine the material to the belt, and if the shoot is correctly designed with this object in view, the required length should not be excessive.

In Fig. 11, the length is shewn as 7′-0″, but where the loading conditions are correctly designed this length can be reduced to as much as 3′-0″ to 4′-0″, as the material would be completely at rest before reaching the end of the skirt boards.

In Fig. 23, is shewn a typical cross-section of wooden skirt boards, fitted at the lower edge with belt guards. These belt guards are either made of leather or a piece of rubber belting held in position to the skirt board or plate by means of attachment bolts and mild steel longitudinal clamping flat. The main object of these strips is to close the space between the belt and the skirting thus preventing the fine material scattering and large pieces from jamming under the skirt boards. The guard strips should not touch the conveyor belt, as if it is allowed to drag on the conveyor, it will naturally cause abrasion.

Referring to Fig. 23, the width of the skirt boards should be as given in Table No. 15.

The height of the skirt boards should be arranged to suit the size of the largest pieces handled and generally they are from 6″ to 12″ high.

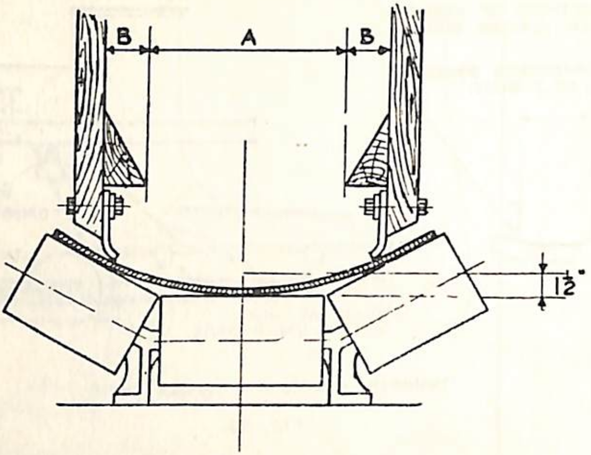


Fig. 23.

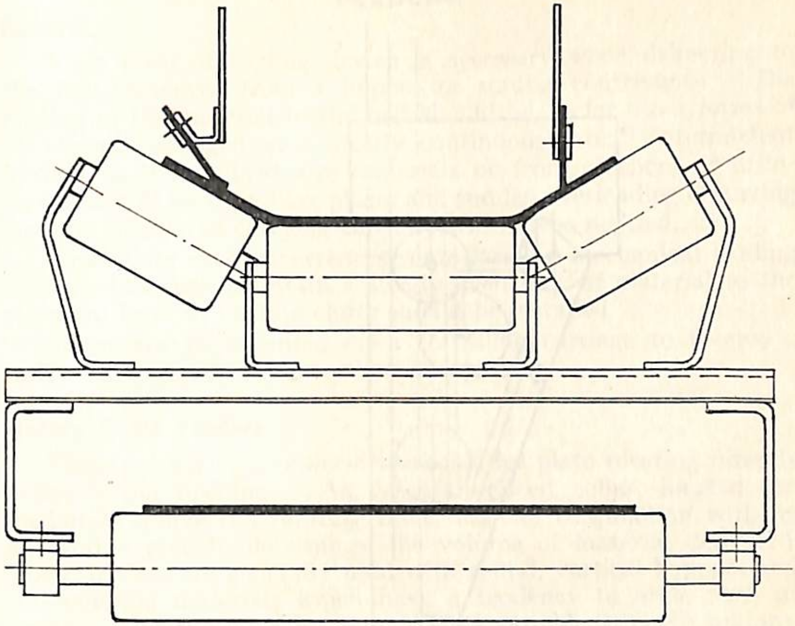
TABLE No. 15.

WIDTH OF SKIRT BOARDS.

Width of Belt. inches.	A	B	C
14	9	6	2⅜
16	11	6	2⅜
18	12	6	2⅜
20	14	6	2⅜
24	17	7	3¼
30	22	9	3¾
36	26	11	4
42	30	13	4¼
48	35	15	5
54	38	18	5¼
60	44	20	6⅛

With steel skirt plates, the deflecting pieces, as shewn in Fig. 24, are not generally adopted, the plates forming a straight line. Steel skirts are usually arranged about two-thirds of the width of the belt and in many instances, such as with materials not liable to cause dust, the guard strips are dispensed with, as the margin of belt at each side is sufficient to prevent spillage.

When feeding an inclined belt, particularly belts inclined above 10°, the material does not so readily acquire a velocity approximating to that of the belt, and skirt boards are particularly



TWO TYPES OF SKIRT PLATES

Fig. 24.

necessary. Due to lumpy material having a tendency to roll back, the length is generally longer than that used for horizontal belts.

Where the loading point is on a short inclined run, depressed to suit the travel of the throw-off carriage, as shewn in Fig. 25, the skirt boards, to act as a precaution against spillage at the bend pulley, are continued the full length of the inclined run. The skirt boards should be made continuous, as intermediate skirt boards placed at the bend or at several points along the run of a horizontal belt where loading takes place are liable to intercept the flow of the material on the belt and wedging or jamming will occur. At the bend pulley the material will automatically open out as the belt flattens and regain its original cross section after leaving same, and belts loaded to the correct carrying capacity will negotiate these bends without spillage.

Only in very rare cases are continuous skirt boards necessary, that is where several feed chutes are arranged along the conveyor, such as is used on conveyors taking material from the various bunker compartments of various sized material, such as sand and gravel plants. Otherwise where such are required the belt has not been designed to suit the largest pieces handled or the feeding arrangement is not satisfactory.

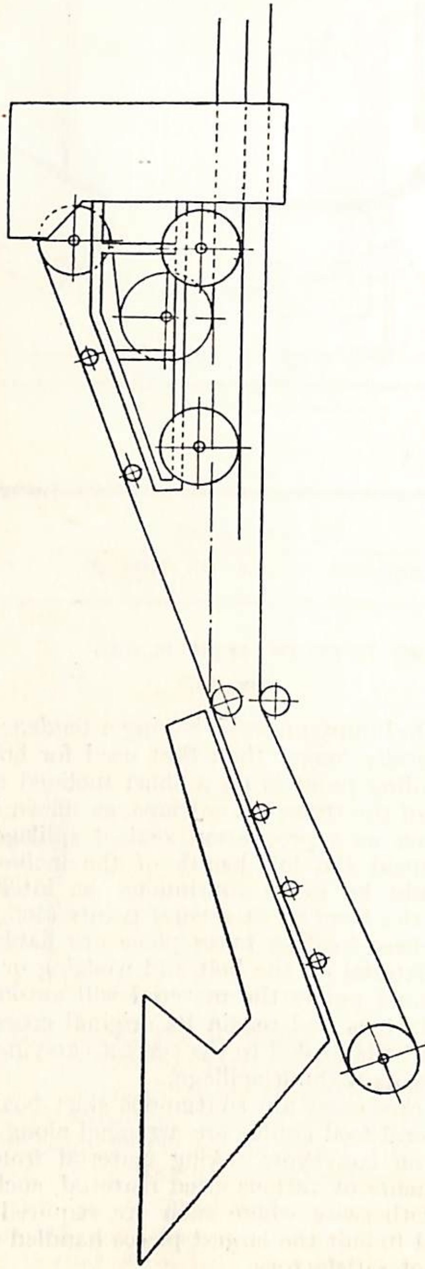


Fig. 25.

FEEDERS.

General.

Some form of feeding device is necessary when delivering to the belt conveyor from a hopper or similar contrivance. The feeding of the material to the belt should be, as for other forms of conveyors, arranged at a steady continuous rate. Intermittent feeding as with mixed size materials or from crushers or disintegrators will result in bare places and sudden overloading occurring and the calculated carrying capacity will not be realised.

To obviate such occurrences some form of mechanical loading device which will maintain a steady even flow of material to the standard form of loading chute should be installed.

These can be mounted on a travelling carriage to receive a feed of material from more than one point.

Rotary Table Feeders.

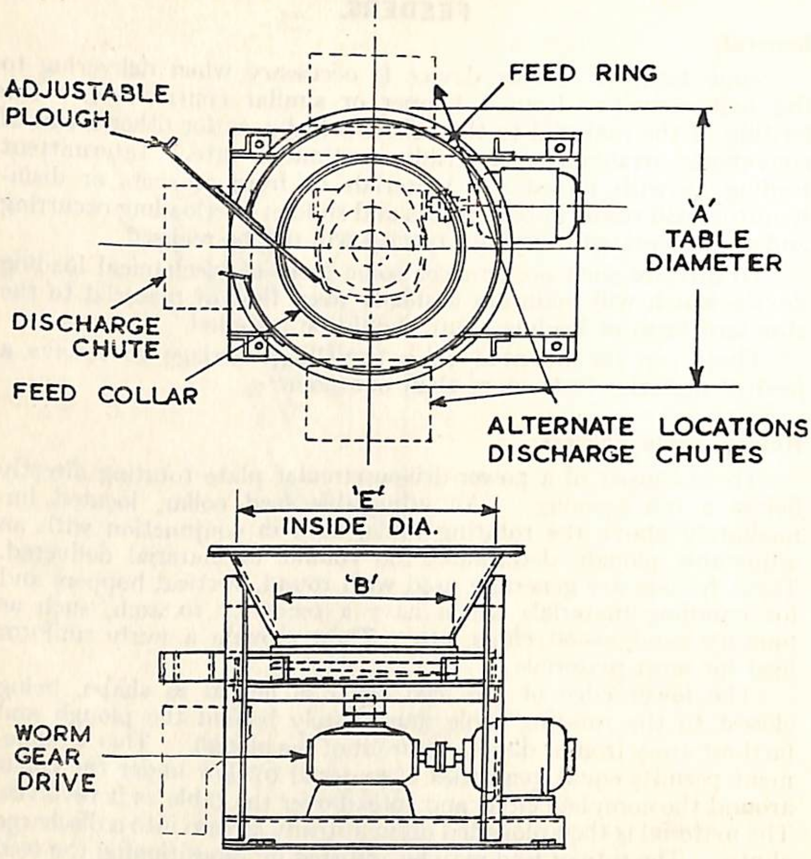
These consist of a power-driven circular plate rotating directly below a bin opening. An adjustable feed collar, located immediately above the rotating table, used in conjunction with an adjustable plough, determines the volume of material delivered. These feeders are generally used with round, vertical hoppers and for handling materials which have a tendency to arch, such as foundry sand, wood chips, etc. They provide a fairly uniform feed for most materials.

The lower edge of the feed collar is helical in shape, being closest to the rotating table immediately behind the plough and farthest away from it directly in front of the plough. This arrangement permits equal quantities of material to flow under the collar around the complete circle and spread over the table as it revolves. The material is then ploughed off in a steady stream into a discharge chute. The rate of feed may be adjusted by repositioning the feed collar and the plough or both.

The size of these feeders range from 24″ to 84″ in diameter, for application to hoppers of like proportion. Because of the relatively large table diameter, hopper openings, which are large in proportion to the hopper diameter, act to reduce arching of the material without withdrawing unduly large quantities.

This type of feeder is readily adapted to blending materials simultaneously while feeding. Various materials can be drawn from a compartmented hopper or from a series of hoppers, the material being directly above the other on the belt, and the rate of flow or mixing can be accurately regulated by the adjustment of each feed collar.

Feeders marked "Y," Table No. 16, are usually fitted with a set of machine cut spur gears between worm box and vertical shaft of feeder.



ROTARY FEEDER

Fig. 26.

Example.—Select a rotary table feeder to suit the following conditions :—

Material	...	foundry sand.
Weight	...	100 lbs. per cubic foot.
Capacity	...	50 tons per hour.

Solution.—Referring to Table No. 16, we find that a 60″ diameter table feeder operating at 6 r.p.m. will handle material weighing 100 lbs. cubic foot at the maximum rate of 54 tons per hour. This size is therefore satisfactory.

TABLE No. 16.

Table Dia. "A" inches.	Capacity cu. ft/hr.		Capacity tons/hr.				
	Min.	Max.	Min.	Max.	H.P.	B.	E.
26 "X"	36	254	1·6	11·3	1	16¾	27¾
36 "X"	50	356	2·2	15·8	1	24	35
48 "Y"	90	834	4·0	37·1	2	28	39
60 "Y"	124	1210	5·5	53·8	3	40	52
72 "Y"	164	1580	7·3	70·0	5	52	64
84 "Y"	186	1850	8·2	82·0	7½	64	76

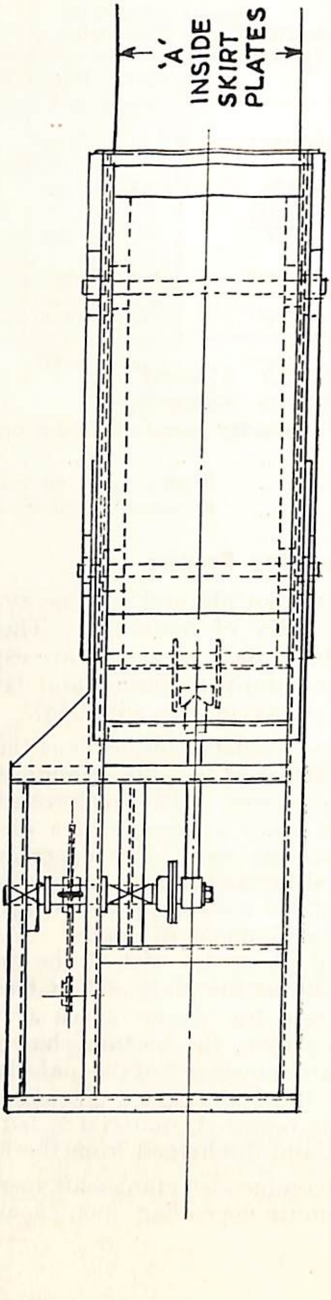
"X" Capacity based on feeder operating speed of 10·8 r.p.m.
"Y" ,, ,, ,, ,, ,, ,, 6 ,,
,, is also based on material weighing 100 lbs. per cu/ft.
,, is directly proportional to weight.

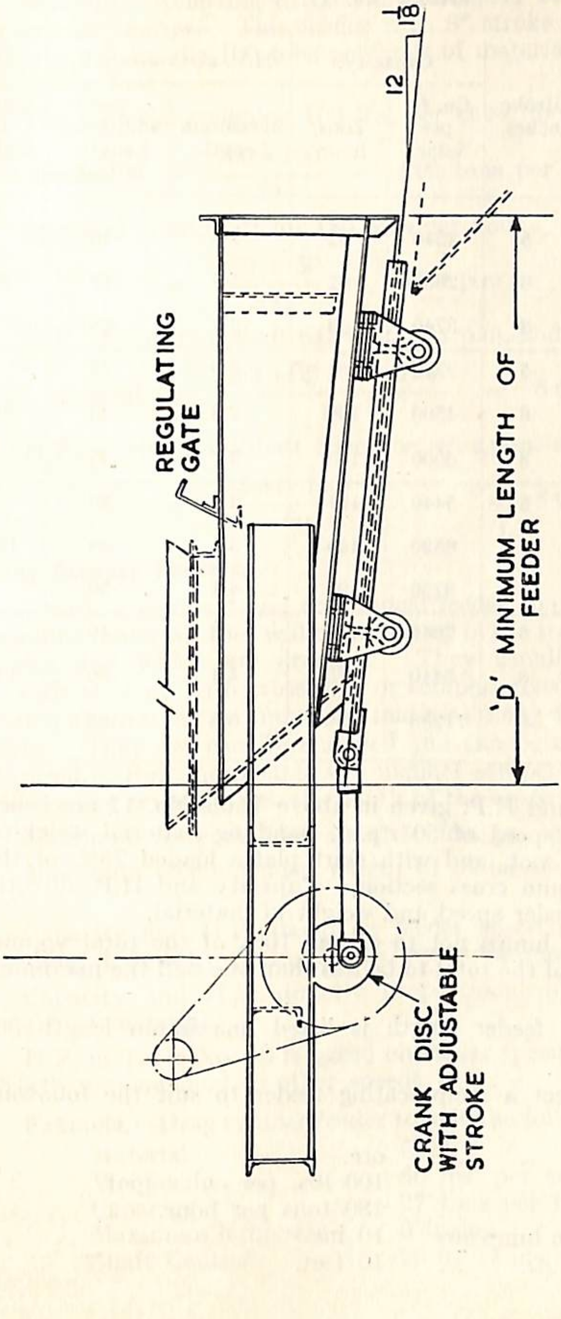
Reciprocating Feeders.

This is an old and reliable type of feeder, used for handling a wide variety of materials. They are widely used for handling coal, stone, ores, sand and are especially adapted for the handling of large lumpy materials and large capacities and material that is non-sticky or non-adhering.

These feeders consist essentially of a pair of stationary skirt plates attached to a bin or hopper beneath which a roller supported plate operates. The reciprocating motion is imparted to the bottom plate by means of a disc crank or an eccentric through a connecting rod. The operation of the feeder is as follows:— material flowing from a storage hopper comes to rest on the feeder plate and is restrained from continued flow by the skirt plates and its natural angle of repose. As the reciprocating plate moves forward, it carries with it the material resting upon it. Simultaneously, as the plate and materials move forward, other material flows into the vacant space at the rear of the feeder. On the return stroke, the bottom plate slides beneath the material, since rearward movement of the material is restrained by the new material which filled the space created on the forward stroke. As the cycle is repeated, material is fed from the hopper on the forward stroke and discharged from the feeder on the return stroke.

Recommended crankshaft speeds range from 10 to 60 revolutions per minute depending upon capacity and size of pieces.


'A' INSIDE SKIRT PLATES


1/8
12
REGULATING GATE
'D' MINIMUM LENGTH OF FEEDER
CRANK DISC WITH ADJUSTABLE STROKE

RECIPROCATING FEEDER

Fig. 27.

TABLE No. 17.

Feeder width in inches. A	Max. Lump Size in inches.	Stroke inches.	Capacity.		H.P. at Crankshaft.		Length in inches.
			Cu. ft. per hour.	Tons/ hour.	Maximum Length.	per additional Length in feet.	
18	6	5	2340	52	1·3	·16	89
		6	2800	62	1·5	·19	
		8	3740	83	2·0	·25	
24	10	5	3780	84	2·0	·22	96
		6	4500	100	2·3	·26	
		8	6000	133	3·1	·35	
30	12	5	5440	121	2·5	·30	101
		6	6520	145	3·1	·38	
		8	8760	195	4·1	·50	
36	15	5	7040	146	3·5	·40	115
		6	8440	188	4·1	·50	
		8	11280	250	5·5	·65	

The capacity and H.P. given in above Table No. 17 are based on a crank shaft speed of 50 r.p.m. handling material weighing 50 lbs. per cubic foot, and with skirt plates loaded 75% of the theoretical maximum cross section. Capacity and H.P. directly proportional to feeder speed and weight of material.

Maximum size lumps not to exceed 10% of the total volume and at least 75% of the total to be less than one-half the maximum lump size.

The minimum feeder length is listed, maximum length 36″ greater.

Example.—Select a reciprocating feeder to suit the following conditions :—

Material ... ... ore.
Weight ... ... 100 lbs. per cubic foot.
Capacity ... 180 tons per hour.
Maximum lump size 10 ins.
Length ... ... 10 feet.

Solution.—Referring to the table, a feeder 24″ wide is required to suit 10″ lumps. This feeder with 6″ stroke and 50″ revs. per minute will handle 100 tons per hour of material weighing 50 lbs. per cubic foot.

The material weighing 100 lbs. per cubic foot, the capacity of

$$\text{the feeder} = \frac{100 \times 100}{50} = 200 \text{ tons per hour.}$$

Speed of crankshaft for 180 tons per hour

$$= \frac{180 \times 50}{200} = 45 \text{ r.p.m.}$$

H.P. at the crankshaft based on 50 r.p.m. and 100 lbs. per cubic

$$\text{foot material} = \frac{2{\cdot}3 + (2)\ ({\cdot}26)\ 100}{50} = 5{\cdot}64 \text{ h.p.}$$

H.P. at the crankshaft based on 45 r.p.m. and same material

$$= \frac{(5{\cdot}64)\ (45)}{50} = 5{\cdot}07 \text{ h.p.}$$

Drag Scraper Feeders.

This is a compact and economical feeder conveyor, suitable for handling materials that will not build up in the trough and conveyor chains and which are granular. They usually consist of two strands of chain with cross bars or scrapers fixed between the two chains at intervals to move the material along the trough bottom plate. They are usually enclosed and can be made dust tight if required. Both horizontal and inclines scraper feeders up to 10° will convey several times the depth of the cross flights or scrapers.

Maximum recommended speed 50 feet per minute.

Maximum recommended weight of material 100 lbs. per cubic foot.

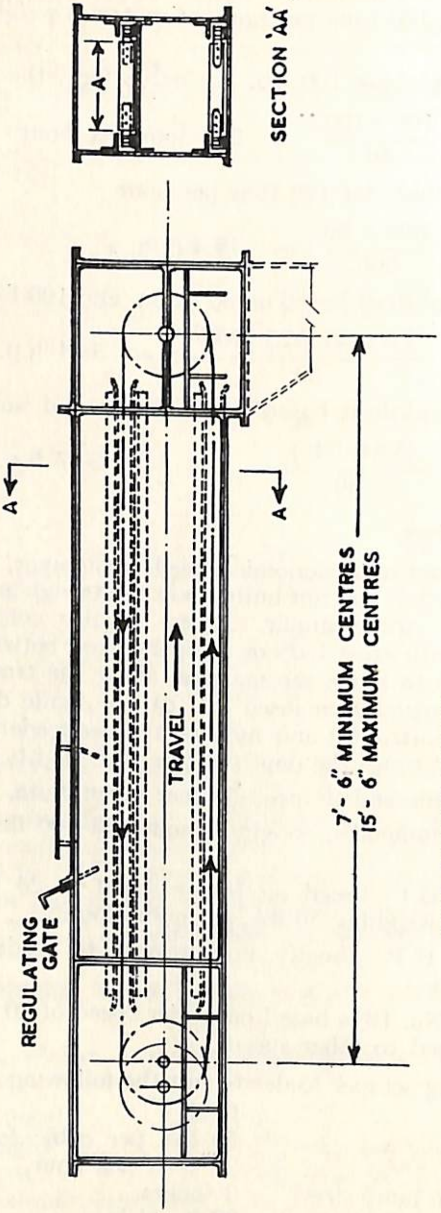
Capacity and H.P. based on feeder speed of 20 ft/min. and handling material weighing 50 lbs. per cubic foot.

Capacity and H.P. directly proportional to feeder speed of weight of material.

H.P. in Table No. 18 is based on feeder speed of 20 ft/min. and directly proportional to other speeds.

Example.—Drag scraper feeder to suit the following conditions :

Material	Coal.
Weight	50 lbs. per cubic foot.
Capacity	27 tons per hour.
Maximum lump size	3 inches.
Shaft Centres ...	26 ft. $1\frac{1}{2}$ ins.



DRAG SCRAPER FEEDER

Fig. 28.

TABLE No. 18.

Width A inches.	Maximum Lump Sizes inches.	Capacity.		H.P. at Headshaft.			Head Shaft Speed R.P.M.	B inches.
		Cu/ft. per hour.	Tons per hour.	Max. Shaft Centres.	Addition Centres per foot.	Lift per foot.		
18	3	728	16·2	·41	·025	·024	5	18
24	3	968	21·5	·52	·032	·027	5	18
36	3	1460	32·4	·97	·042	·040	5	24
18	4	728	16·2	·60	·035	—	5	—
24	4	968	21·5	·80	·045	—	5	—

Solution.—Referring to table, an 18″ wide feeder is satisfactory for the lump size and operating at 20 ft. per minute will handle 16·2 tons per hour of material weighing 50 lbs/cubic foot.

The speed of the feeder for a capacity of 27 tons per hour

$$= \frac{27 \times 20}{16{\cdot}2} = 33 \text{ feet per minute.}$$

The H.P. at the headshaft based on 20 feet per minute

$$= {\cdot}60 + (11{\cdot}5)\ ({\cdot}035) = 1 \text{ h.p.}$$

The H.P. at the headshaft based on 33 feet per minute

$$= \frac{1 \times 33}{20} = 1{\cdot}65 \text{ h.p.}$$

The speed of the headshaft for 33 feet per minute

$$= \frac{5 \times 33}{20} = 8{\cdot}25 \text{ r.p.m.}$$

Screw Feeders.

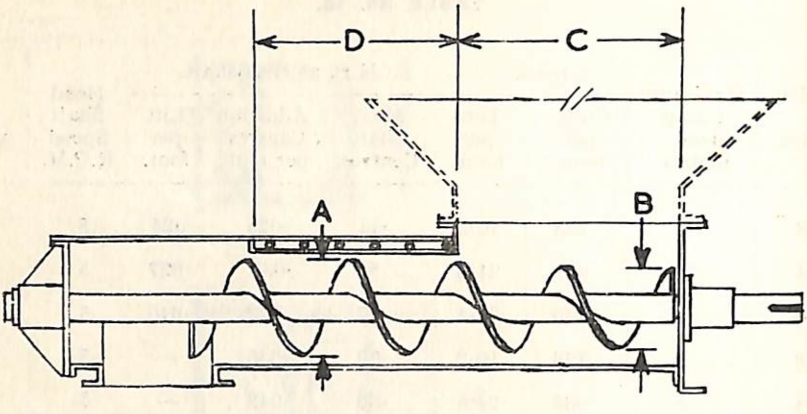
These are short screw conveyors used to control the flow of material at a constant or variable rate from receiving or storage hoppers, etc. They are suitable for handling a wide variety of materials ranging from fines to a combination of fines and lumps.

They are totally enclosed, compact, simple in design, and can be dust tight in construction.

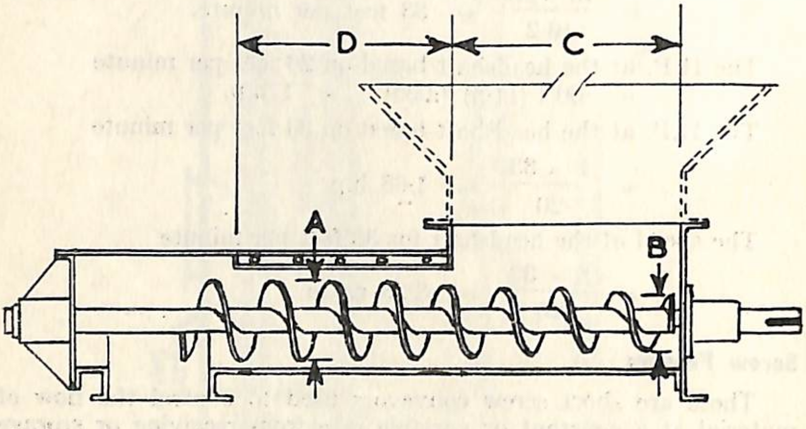
There are two general types of screw feeders :—

(1) Consisting of a tapered screw with regular pitch segments.

(2) A tapered screw with short pitch segments.



TYPE 'A' FEEDER



TYPE 'B' FEEDER

SCREW FEEDERS

Fig. 29.

It is necessary to have a tapered screw to handle material containing a fair percentage of lumps and also to draw the material uniformly across the entire width of the feed opening.

When it is necessary to convey the material over a distance, a screw conveyor can be combined with the feeder to make the extension.

It is necessary to consider the following factors when considering a screw feeder :—

Kind and character of material to be handled.

Weight of material in lbs. per cubic foot.

Maximum rate in tons per hour, or cubic feet per hour.

Maximum size lumps and average size of material.

The capacity of screw feeders can be obtained from Table No. 19 after determining that the material can be satisfactorily handled in a screw feeder.

The H.P. required to drive a screw feeder depends largely upon the material being handled. The following formula may be used for calculating the H.P. :—

$$\text{H.P.} = \frac{A\,(L + (BC)\,K}{1{,}000{,}000}$$

Where A = Capacity in cubic foot per hour.
L = Total length of feeder in feet.
C = Exposed length of screw under inlet in feet.
B = Constant = 2 for material 0 to $\frac{1}{8}''$.
= 3 for material Granular $\frac{1}{8}''$ to $\frac{1}{2}''$ and for material containing lumps $\frac{1}{2}''$ and over.
K = H.P. factor as given in Table No. 20.

The maximum lump sizes listed should not exceed 25% of the total volume.

Example.—Select screw feeder to suit the following :—

Material	Cotton seed meal.
Weight	35-40 lbs. per cubic foot.
Capacity	600 cubic foot per hour. $10\frac{1}{2}$ tons per hour.
Maximum lump size	$\frac{1}{8}''$. 10% total volume. Length of feeder. 7 feet.

Solution.—Use Type A feeder.

Referring to capacity table, a 9″ diameter feeder operating at 50 r.p.m. will handle 625 cubic foot per hour, since the capacity

TABLE No. 19.

Screw Diameter inches.	Maximum Lumps inches.	Approximate Capacity Cubic ft/hr. Speed of Screw R.P.M.										A	B	C	D
		10	20	30	40	50	60	70	80	90	100	inches.			
Type A.															
6	⅛	45	90	135	180	225	270	315	360	405	450	6	4	10	10
9	⅛	145	290	435	580	725	870	1015	1160	1305	1450	9	6	16	16
12	⅛	375	750	1125	1500	1875	2250	2625	3000	3375	3750	12	9	24	24
14	⅛	580	1160	1740	2320	2900	3480	4060	4640	5290	5870	14	10	28	28
16	⅛	870	1740	2610	3480	4350	5220	6090	6960	7830	8700	16	12	32	32
Type B.															
6	¾	30	60	90	120	150	180	210	240	270	300	6	4	12	12
9	1½	100	200	300	400	500	600	700	800	900	1000	9	6	24	24
12	2	250	500	750	1000	1250	1500	1750	2000	2250	2500	12	9	36	36
14	2½	380	760	1140	1520	1900	2280	2660	3040	3420	3800	14	10	42	42
16	3	550	1100	1650	2200	2750	3300	3850	4400	4950	5500	16	12	48	48

is slightly greater than required the speed can be decreased if desired as follows :—

$$\frac{625 \times 600}{50} = 48 \text{ r.p.m.}$$

$$\text{The H.P.} = \frac{600\ [7 + (3)\ (2)\ ]\ 29}{1{,}000{,}000} = \cdot 23 \text{ at Feeder shaft.}$$

Apron Feeders.

This type of feeder consists of an endless apron, operating over head and foot terminals with suitable intermediate supports.

TABLE No. 20.

Material Class.	Weight of Material lbs. cu/ft.	H.P. Factor K.						
		Screw diameter in inches.						
		6	9	12	14	16	18	20
Fine free flowing materials. Beans, Rice, Corn, Seeds, etc.	10	17	10	9	8	8	8	7
	14	19	12	10	10	9	9	9
	19	21	14	12	12	11	11	11
	27	24	17	15	15	15	14	14
	38	28	21	20	19	19	19	18
	53	45	38	37	36	36	35	35
	74	64	57	55	55	54	54	53
Semi-free flowing Copra, Coal, Salt.	5	16	9	8	7	7	7	6
	7	17	10	9	8	8	8	7
	10	19	12	11	10	10	10	9
	14	22	15	13	13	12	12	11
	19	25	18	16	16	15	15	14
	27	29	22	21	20	20	20	19
	38	36	29	27	27	27	26	26
	53	56	49	47	47	46	46	45
	74	72	65	64	63	63	63	62
	104	138	131	129	129	129	128	128
Such Materials Brewers Grain, Hops Spent Dry and wet, Malt wet and dry, Fine oxide	5	18	11	10	9	9	9	8
	7	20	13	12	11	11	11	10
	10	23	16	15	14	14	14	13
	14	27	20	19	18	18	18	17
	19	32	25	24	23	23	23	22
	27	40	33	32	31	31	31	30
	38	51	44	43	42	42	42	41
	53	77	70	68	68	67	67	66
	74	102	95	93	93	92	92	91
	104	180	173	171	171	170	170	169

The apron or trough consists of two strands of chain to which a series of overlapping pans or joggled plates are attached to carry the material to be handled. The sides of various designs are attached to prevent spillage. Stationary skirt plates, or specially constructed hopper outlets serve to confine the material and provide the necessary depth of bed for desired capacity or to accommodate large lumps. They find their chief use for materials difficult to handle due to being abrasive, heavy or lumpy, such as run of mine coal, stone ore, etc. (Fig. 30).

Capacity and H.P. (Table No. 21, page 72) is based on feeder speed of 20 ft/mins. and handling material weighing 100 lbs. per cubic foot and skirt plates loaded 75% of theoretical maximum cross section.

Capacity and H.P. directly proportional to feeder speed and weight of material. Recommended minimum height of skirts 1½ times maximum lump size. Maximum lump size for unsixed material not to exceed 10% of total volume and at least 75% of the total to be less than ½ the maximum size lump. (Table No. 21).

Belt Feeders.

These are similar in construction to belt conveyors and consist of endless belts operating over supporting idlers between head and foot pulleys. Conveyor belts having a heavy rubber cover on the carrying side are used for carrying the material. Closely spaced flat belt idlers provide the necessary support for the belt and material. Fixed skirt plates are provided the full length of the feeder to confine the material, and rubber sealing strips are also employed along the lower edge of the skirt plates to reduce belt clearance and spillage.

Maximum angle of incline 10 degrees.

,, recommended speed 60 ft/min. for abrasive materials.

Maximum recommended speed 100 ft/min. for non-abrasive materials.

Maximum recommended weight of material 100 lbs/cu. ft.

Capacity and H.P. based on feeder speed of 20 ft/min. handling material weighing 50 lbs/cu. ft. and with skirt plates loaded 75% theoretical maximum cross section.

Capacity and H.P. directly proportional to feeder speed and weight of material.

Maximum size lumps not to exceed 10% of the total volume and at least 75% of the total to be less than half the maximum lump size.

Maximum centres 18 feet. (Table No. 22).

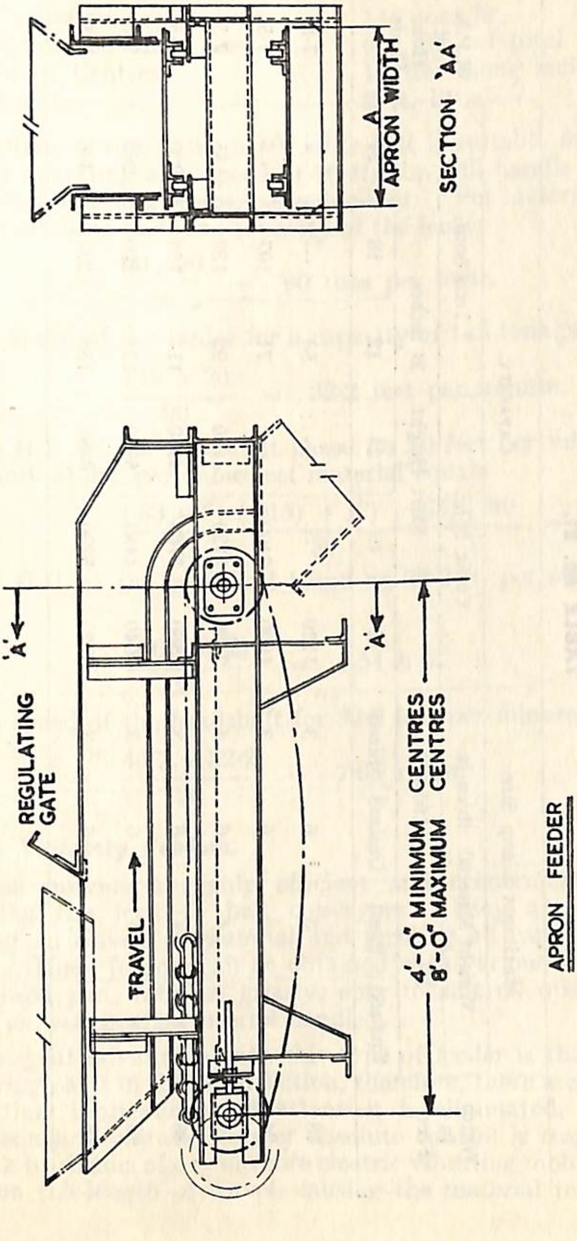

APRON WIDTH
SECTION 'AA'
REGULATING GATE
'A'
'A'
TRAVEL
4'-0" MINIMUM CENTRES
8'-0" MAXIMUM CENTRES
APRON FEEDER

Fig. 30.

TABLE No. 21.

Width in Feet.	Maximum Lump Size.				CAPACITY.						R.P.M. of Head Shaft.
	Weight Material. lb/cu/ft.				Cubic ft/hr.			Tons/hour.			
	50		100/125		Skirt Height. B inches.						
	Unsized	Sized	Unsized	Sized	12	18	24	12	18	24	
18	6	3	6	3	1120	—	—	50	—	—	7·5
24	8	4	6	3	1600	2400	—	71	107	—	7·5
30	8	4	6	3	2080	3120	4160	92	139	185	7·5
36	8	4	6	3	2560	3760	4960	114	168	220	7·5
42	8	4	6	3	3040	4480	5920	135	200	263	7·5
48	8	4	6	3	3520	5200	6880	156	231	306	7·5

Example.—Select a belt feeder to suit the following details :—

Material	Ore.
Weight	90 lbs/cu. ft.
Capacity	145 tons/hr.
Maximum lump size ...	6″. 8% of total volume.
Shaft Centres	10 feet along incline.
Incline	2 ft. lift.

Solution.—From table a 30″ wide belt is suitable for the lump size and operating at a speed of 20 ft/min. will handle 50 tons/hr. of material weighing 50 lbs. per cubic foot. For material weighing 90 lbs. per cubic foot, the capacity of the feeder

$$= \frac{50 \times 90}{50} = 90 \text{ tons per hour.}$$

The speed of the feeder for a capacity of 145 tons per hour

$$= \frac{145 \times 20}{90} = 32{\cdot}2 \text{ feet per minute.}$$

The H.P. at the headshaft based on 20 feet per minute feeder speed and 90 lbs. per cubic feet material equals

$$\frac{[{\cdot}33 + (5)\ ({\cdot}015)\ +\ (2)\ ({\cdot}065)\ ]\ 90}{50} = {\cdot}96 \text{ h.p.}$$

The H.P. at the headshaft based on 32·2 ft. per minute feeder speed

$$= \frac{({\cdot}96)\ (32{\cdot}2)}{20} = 1{\cdot}54 \text{ h.p.}$$

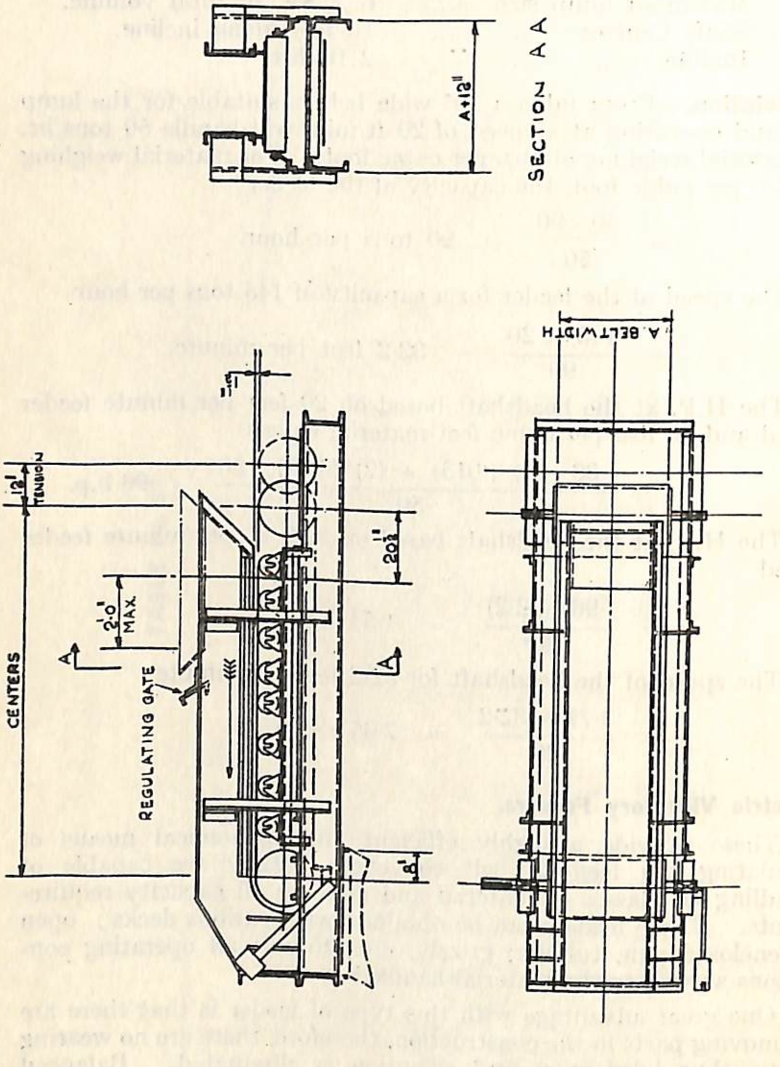
The speed of the headshaft for 32·2 feet per minute

$$= \frac{4{\cdot}77 \times 32{\cdot}2}{20} = 7{\cdot}65 \text{ r.p.m.}$$

Electric Vibratory Feeders.

These provide a highly efficient and economical means of regulating the feed to belt conveyors. They are capable of handling all classes of material and meeting all capacity requirements. These feeders can be obtained with various decks ; open or enclosed pan, tubular, grizzly, etc., to suit all operating conditions as well as the material handled.

One great advantage with this type of feeder is that there are no moving parts in the construction, therefore, there are no wearing parts, thus lubrication and attention is eliminated. Balanced high frequency vibration under absolute control is transmitted to the deck by means of one or more electric vibrating motors, depending upon the length of travel, causing the material to move in a



BELT FEEDER

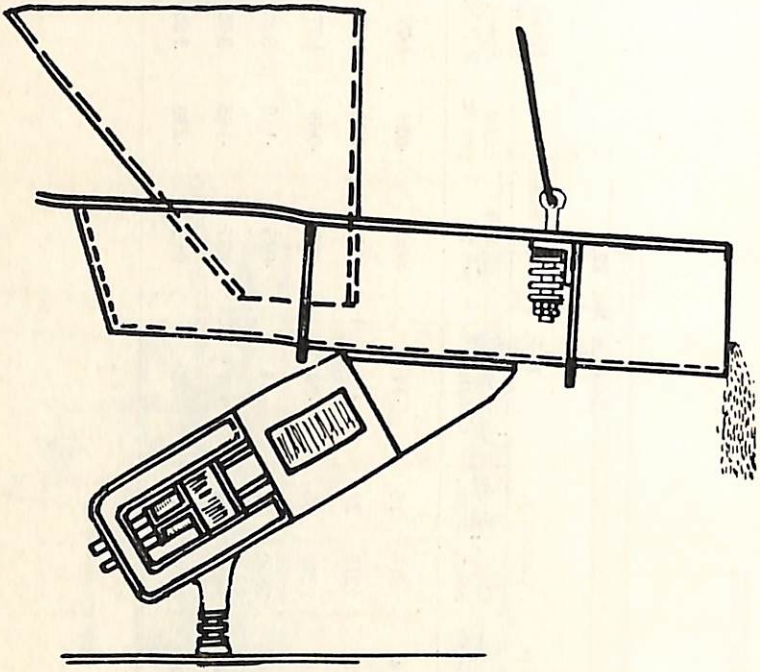
Fig. 31.

TABLE No. 22.

Belt width A	Maximum Lump Size. Inches.		Capacity.		H.P. at Head Shaft.			Head Shaft Speed R.P.N.	Min. Shaft Crs. inches.	Pulley Dia. inches.	B inches.
	Non-Abrasive.	Abrasive.	Cu/ft. hour.	Tons/ hour.	Max. Shaft Centres.	Add. Crs. per ft.	Lift per ft.				
18	4	4	810	18	·14	·008	·025	4·77	54	16	12
24	5	5	1215	27	·20	·010	·035	4·77	54	16	16
30	6	6	2250	50	·33	·015	·065	4·77	60	16	22
36	8	6	3850	82	·44	·025	·110	3·82	66	20	28
42	10	6	5620	125	·57	·037	·165	3·82	72	20	34
48	12	6	8100	180	·76	·055	·230	3·82	78	20	40

continuous uniform flow. Rate of feed can be regulated while the unit is running. Response is positive and instantaneous. No mechanical adjustments are required and power consumption is extremely low.

Fig. 32 shows the application of an electric vibratory feeder, feeding a belt conveyor. They can be operated on the level, upgrade or downgrade and will take feed from either hoppers or bins.



ELECTRO-MAGNETIC VIBRATORY FEEDER

Fig. 32.

CARRYING CAPACITY OF BELT CONVEYORS.

Belt conveyors may be built to handle practically any quantity of most materials. The actual capacity per hour is dependent upon the rate at which the material is supplied or fed to the belt, delays due to intermittent supplies or when fed direct from a crusher or grab and similar loading conditions are liable to cause interruptions in the supply, consequently reducing the carrying rate. It is therefore necessary to consider the carrying capacity under normal working conditions that is when full loading is being maintained at peak load. The usual rules given for the carrying capacity of belts are generally based on a continuous uniform flow of material to the belt throughout the unit of time specified and as previously stated, care should be taken that the quantity of material required to be carried can be fed and properly carried by the belt under the operating conditions imposed.

When the quantity to be conveyed is small and the pieces large, the maximum size of the latter determines the width of the belt, and the speed should be no faster than that which is necessary to carry the desired load comfortably.

When, on the other hand, the quantity is great and the pieces small, the capacity determines the width of the belt. A conveyor running at a slow speed may be loaded more deeply than one at a high speed, and when a narrow belt is run much above the advisable speed, the load thins out and the capacity does not increase with the speed.

No definite carrying capacity can be fixed for belts and each case should be considered on its own merits, having regard to the class and nature of the material being handled, speed of the belt and the type of feeding device adopted.

There is also the question of spillage to consider, and if a belt is carrying large pieces over a fines hopper, it may be necessary to avoid spillage at all costs, in which case the ordinary carrying capacity may require to be reduced to insure against this possibility.

Knowing the weight per cubic foot and size and percentage of lumps in the material to be handled, turn to the capacity table No. 23. Select a combination of belt width and speed that will handle the lumps and convey the peak load. The figures given are for troughed belts carried on three pulley idler sets having the outer inclined rollers at 20° ; capacities for flat conveyors will be 50% less. If the unit desired lies between two figures given, interpolate for the proper value. If the belt is not uniformly loaded, the capacity, power and belt strength must be figured for peak load.

When several combinations of belt speed and width will handle the load, the narrowest belt operated at the highest speed will

TABLE No. 23.
CAPACITIES OF TROUGHED BELT CONVEYORS.
20° IDLERS.

Weight of material lbs/cu/ft.	Belt width inches.	Capacity Tons Per Hour.						Maximum Size Lumps Inches.	
		Belt speed. Feet per minute.						If uniform in size.	If mixed with fines.
		100	200	300	400	500	600		
	12	6·2	12·4	18·6	—	—	—	2	3
	14	8·0	16·0	24·0	—	—	—	2	3
	16	11·2	22·4	33·6	—	—	—	3	5
	18	14·2	28·4	42·6	57	—	—	3	5
	20	17·6	35·2	52·8	71	—	—	4	6
30	24	25·7	51·5	77·2	103	129	—	4½	8
Coke	30	42	84	126	168	209	—	6	11
	36	62	124	186	248	311	372	8	15
	42	87	174	261	348	435	523	10	18
	48	117	234	351	468	585	702	12	21
	54	151	302	453	604	755	906	14	24
	60	193	386	578	772	965	1155	16	28
	12	10·3	20·6	31·0	—	—	—	2	3
	14	14·0	28·0	42·0	—	—	—	2	3
	16	18·6	37·2	56	—	—	—	3	5
	18	23	47	71	94	—	—	3	5
	20	30	60	90	120	—	—	4	6
50	24	44	88	132	175	218	—	4½	8
Coal	30	70	140	210	280	350	—	6	11
	36	103	206	310	413	516	620	8	15
	42	145	290	435	580	725	870	10	18
	48	195	390	583	778	973	1165	12	21
	54	253	506	760	1013	1266	1520	14	24
	60	320	640	960	1280	1600	1920	16	28
	12	15	30	46	—	—	—	2	3
75	14	21	42	64	—	—	—	2	3
Loose	16	28	56	84	—	—	—	3	5
Earth	18	38	76	114	152	—	—	3	5
	20	45	90	135	180	—	—	4	6
	24	65	130	195	260	325	—	4½	8

TABLE No. 23—contd.
CAPACITIES OF TROUGHED BELT CONVEYORS.
20° IDLERS.

Weight of material lbs/cu/ft.	Belt width inches.	CAPACITY TONS PER HOUR.						MAXIMUM SIZE LUMPS INCHES.	
		Belt speed. Feet per minute.						If uniform in size.	If mixed with fines.
		100	200	300	400	500	600		
	30	105	210	315	420	525	—	6	11
75	36	155	310	465	620	775	930	8	15
Loose	42	214	428	642	856	1070	1284	10	18
Earth	48	292	584	876	1168	1460	1750	12	21
	54	380	760	1140	1520	1900	2280	14	24
	60	480	960	1440	1920	2400	2880	16	28
	12	20	41	63	—	—	—	2	3
	14	28	56	84	—	—	—	2	3
	16	37	74	112	—	—	—	3	5
	18	46	74	142	188	—	—	3	5
100	20	60	120	180	240	—	—	4	6
Sand	24	88	176	264	350	432	—	4½	8
Gravel	30	140	280	420	560	700	—	6	11
	36	206	412	620	825	1030	1240	8	15
	42	290	580	870	1160	1450	1740	10	18
	48	390	780	1165	1560	1950	2330	12	21
	54	506	1012	1520	2025	2530	3040	14	24
	60	640	1280	1920	2560	—	—	16	28
	12	31	62	93	—	—	—	2	3
	14	42	84	126	—	—	—	2	3
	16	56	112	168	—	—	—	3	5
	18	70	140	210	280	—	—	3	5
	20	90	180	270	360	—	—	4	6
150	24	131	262	393	524	655	—	4½	8
Ore	30	210	420	630	840	1050	—	6	11
	36	310	620	930	1240	1550	1860	8	15
	42	435	870	1305	1740	2175	2610	10	18
	48	585	1170	1755	2340	2925	—	12	21
	54	760	1520	2280	3040	—	—	14	24
	60	960	1920	2880	—	—	—	16	28

usually be the most economical. If the material is of a fragile nature or includes large lumps a wider belt and slower speed may be advisable.

Example.—Find the width and speed of a belt conveyor to handle 360 tons of coal per hour, the coal weighing 50 lbs. per cubic foot and containing lumps 12″ cube. Table No. 23, for material weighing 50 lbs. per foot, gives the following combinations :

36″ Belt at 350 f.p.m.
42″ ,, ,, 250 ,,
48″ ,, ,, 185 ,,

Of these combinations we find that all will carry 12″ lumps, but the 42″ wide belt at 250 feet per minute will minimize the breakage of the coal and still give the most economical installation.

The capacities given in Table No. 23 are based on uniformly loaded belts, using the Goodyear revised formula,

$$T = \frac{(W + 180)\ W^2\ S\ M}{12{,}000{,}000} \times \cdot 89$$

in which T = Peak capacity in tons per hour.
W = Width of belt in inches.
S = Speed of belt in feet per minute.
M = Weight of material in lbs. per cubic foot.

This formula takes into consideration the fact that wide belts will handle proportionately greater volumes than narrow belts without spillage.

Capacities vary as shown in Table 24 and varies from $3{\cdot}2\ W^2$ cubic feet for 12″ wide belts up to $4{\cdot}0W^2$ cubic feet per hour for 60″ wide belts at a speed of 100 feet per minute. While the formula given and the Table No. 23 give a maximum capacity practical for general use, it is possible to carry as much as 25% greater volume only under the following ideal conditions.

1. Equipment must be well-designed and maintained.
2. Incline of conveyor not more than a few degrees from horizontal.
3. Load chute and loading conditions are practically perfect.
4. Belt loaded on an horizontal run.
5. Material not to include large lumps or contain a high percentage of lumps of any kind.

The following Table, No. 24, gives the cross sectional area of load, in square feet, of the load handled by troughed conveyor belts as obtained by using the foregoing formula.

TABLE No. 24.

CROSS SECTIONAL AREA.

Width of Belt in inches.	Cross Sectional Area of Load in square feet.	Width of Belt in inches.	Cross Sectional Area of Load in square feet.
12	·077	36	·778
14	·108	42	1·088
18	·178	48	1·458
24	·326	45	1·896
30	·525	60	2·400

Flat Belt Conveyors Handling Bulk Material.

These should be figured at 50% of that given for standard troughed belts. For all practical purposes the speed and horse power data given for troughed belts can be used.

Flat Belt Conveyors Handling Packaged Material.

Belt conveyors are often used for carrying packages, bags, boxes, etc., in manufacturing, shipping and storage departments, for processing, assembling orders, or simply for transportation. Troughed belts may be used especially in handling bagged material: however, flat belts are more frequently used.

The capacity and size of flat belt conveyors are determined by the size, weight and number of pieces to be handled. The capacity is further limited by the number of pieces that can be handled at the loading and discharge points. The most common speeds are from 50 to 100 feet per minute. While the number of plies of the belt are determined to satisfy the required maximum tension, a minimum of four plies is recommended except for extremely light service on narrow belt conveyors of short centres where 2 or 3 ply belts may be used. With these conveyors the service is generally of an intermittent nature, as when loading or unloading is done by hand the size and weight of the boxes, bags, parcels, etc., have to be taken into account, also the speed of the conveyor which, as previously stated, is much slower than belt conveyors handling loose bulk material. It is difficult, therefore, to fix definite rating for package conveyor, but in Table 25 is given by Jeffrey Manufacturing Co., U.S.A., what may be reasonably carried by package conveyors for their various types and size of rollers.

TABLE No. 25.

CAPACITIES OF FLAT BELT CONVEYORS FOR PACKAGES, BOXES, BUNDLES, ETC.

Width of Belt in inches.	2-inch Diameter Rollers.			4-inch Diameter Rollers. (Reliance Jr.).			4-inch Diameter Rollers. (Reliance).		
	*Average Uniform Load in lbs. per sq. feet of Belt.	†Maximum Single Load in lbs.	Std. Roller Spacing in feet.	*Average Uniform Load in Lbs. per sq. feet of Belt.	†Maximum Single Load in lbs.	Std. Roller Spacing in feet.	*Average Uniform Load in lbs. per sq. feet of Belt.	†Maximum Single Load in lbs.	Std. Roller Spacing in feet.
14	30	75	4·5	40	90	4·0	100	250	4·0
16	25	75	4·5	35	90	4·0	95	250	4·0
18	25	75	4·0	30	85	4·0	85	250	4·0
20	20	75	4·0	25	85	4·0	75	250	4·0
24	20	75	3·5	20	70	3·5	65	250	3·5
30	15	65	3·5	15	60	3·0	50	200	3·0
36	—	—	—	—	—	—	35	150	3·0
42	—	—	—	—	—	—	25	125	3·0
48	—	—	—	—	—	—	20	100	3·0

* When Maximum single loads are laid flat (not dropped) upon the belt, the maximum load given in Table may be doubled.

† These loads may be increased proportionally as the roller spacing is decreased. For example, if the roller spacing is reduced to one-half the standard spacing, the average uniform load may be doubled, providing that it does not exceed the maximum single load tabulated for a given size.

The capacity in tons per hour may be determined as follows :—

$$C = \frac{W \times 60 \times S}{L \times 2240} \text{ or } C = \frac{\cdot027\ W.S.}{L}$$

Where C = Capacity tons per hour.
L = Centres of head and tail pulleys in feet.
S = Belt Speed in feet per minute.
W = Maximum total load on conveyor from loading point to discharge point.

This type of conveyor can be arranged to carry on both upper and lower runs and frequently the belt is run between skirtboards or rails, short conveyors are also operated with the belt sliding on polished hardwood boards or in steel troughs. Relieving rollers can be used if desired.

Capacity of Grain Belts.

The capacity of belt conveyors handling grain, wheat, cereals, etc., is usually measured in bushels. It is usual now to install troughed belts as against flat belts due to their larger carrying capacity.

The following table gives the capacity of troughed belt conveyors in bushels per hour as 100 feet per minute.

TABLE No. 26.

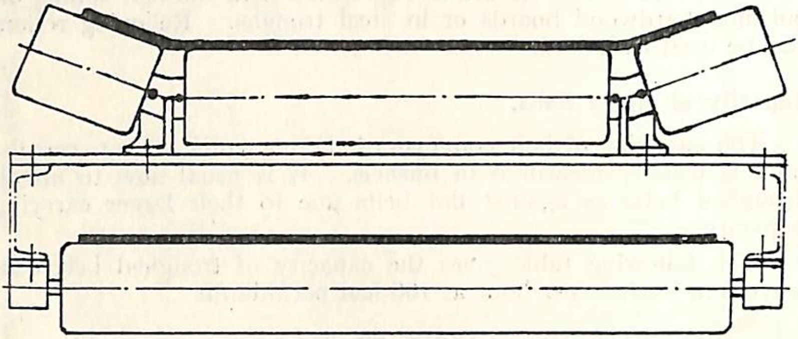
Width of Belt.	Bushels per hour at 100 ft/min.	Width of Belt.	Bushels per hour at 100 ft/min.
12	370·0	42	5217·6
18	854·4	48	7198·4
24	1564·8	54	9100·8
30	2520·0	60	11520·0
36	3734·4		

Carrying Capacity of Picking Belts.

A belt conveyor makes a very effective picking table. It may be either a flat or a troughed belt type. These idlers have a longer centre roll than the standard troughing idlers for straight conveyor service and have short rolls on each side inclined so as to prevent the material from spilling over the edge of the belt.

The capacity of picking and sorting belts is governed by the size and nature of the material handled, the thickness of the load which will readily expose all material for inspection, percentage of refuse to be removed and the number and speed of the pickers or sorters. For bituminous coal the layer is about 6″ deep, which approximates to 10 to 15 tons per hour per foot width of belt. The width of the belt to allow the pickers to easily reach the material should not exceed 48″, 36″ being the usual average and giving the best results.

Fig. 33 shows the usual type of troughing set for picking belts, giving a much broader flat surface than is the case with the ordinary three-pulley carrier set.



PICKING OR SORTING BELT

Fig. 33.

Usually for average conditions the capacity can be figured at 50% of that given in the tables for standard troughed belts.

The usual speed of these belts range from 25 to 50 feet per minute for effective sorting or picking, but may be higher, even up to 100 feet per minute if the material is coarse, comparatively clean and the refuse easily discernable.

SPEEDS.

The speed at which belt conveyors are operated greatly exceeds in ordinary cases that which is obtained with scraper, push-plate or gravity bucket conveyors, due chiefly to the complete separation of the moving parts from the material being conveyed, also noise is practically eliminated.

There are several conditions which have to be taken into consideration before the correct speed for any belt can be determined, such as the width of the belt and also the size and nature of the material. It is therefore not possible to give any fixed figures.

It will be seen from Table 27 that the speed increases as the belt width increases, but it should also be understood that these speeds are maximum figures and the size of the material should also be considered.

TABLE No. 27.

NORMAL AND MAXIMUM RECOMMENDED SPEEDS.

Belt width in ins.	Normal operating speeds in ft/min.	Maximum speeds in feet per minute.		
		Grain and similar light materials.	Coal and similar material.	Heavy Ore, etc.
12	200	400	300	250
14	200	400	300	250
16	200	500	300	250
18	250	500	400	300
20	300	600	400	300
24	300	600	500	350
30	350	700	500	350
36	400	800	600	400
48	400	800	600	400
54	450	800	600	400
60	450	800	600	400

For large lumpy, or mixed fines and large materials of a heavy nature, the speed should not exceed 250 feet per minute, irrespective of the width of the belt, the speed being gradually increased to the maximum as given in relation to the belt width as the size of the material decreases.

There is great difficulty in loading the former class of material at high speeds to obtain the rated capacity, due to the intermittent action of the large pieces on the feeding device or shoot, and consequently the material on the belt will not form a continuous uniform layer. With easy flowing materials like sand, crushed coal, etc., an even layer can be readily obtained at high speeds.

Very often the width as determined from the size of the largest pieces (see width of belts), will be greater than that required to convey the specified or required capacity. In this case, there is no advantage in operating the belt at a speed in excess of that which will give the required capacity.

A fully-loaded belt run at a slow speed carries a deeper layer of material, thus reducing the proportion of the material in contact with the belt and thereby reducing the wear on the cover.

Heavy, sharp, cutting or lumpy materials fed to a fast-running belt will cause cuts to appear and such action soon ends the life of the belt.

When the belts are very short the speed should not exceed that which is required to convey the desired amount, due to the greater wear obtained than is the case with long belts, on account of frequent passing of each foot of belt length past the loading point.

In operating belt conveyors in series, especially if any of the sections are elevating there should be a gradual stepping up of the speed (unless made wider), from the initial receiving conveyor to the ultimate distributing conveyor. This is most essential where it is desired to prevent breakage and the general rule is to increase the speed of each conveyor by 15 feet per minute over the preceding one, the maximum advisable speed as given in Table 27 never being exceeded.

An inclined belt should, to allow correct loading to take place, be operated at a slower speed than a horizontal belt conveyor, but unfortunately this fact is very often overlooked. As previously pointed out, it is not possible to attain the correct velocity of the material in the chute to reduce the abrasive action at the loading point to a minimum, and due to the boiling which takes place at high speeds the belt is subjected to much wear and damage and its life is consequently considerably reduced.

With movable trippers the conveyor is very often arranged with a short inclined run at one end, as shewn in Fig. 25, to allow a longer operating run for the tripper. In this case the loading on the short inclined run is not very frequent and the conveyor could be considered as an ordinary horizontal run when deciding the operating speed, but it is recommended that 300 feet per minute should be a maximum speed.

When ploughs are used the belt speed should not exceed 200 feet per minute.

If the material is wet the approximate minimum speed should be 300 feet per minute.

The speeds for sorting and picking belts, as previously stated, should allow of correct sorting or examination by the attendants, and may vary from 50 feet per minute for coal picking belts to 25 feet per minute for material requiring careful examination.

The speeds of package conveyors are varied to suit the conditions. For small light bags or parcels the speed may be as high as 200 feet per minute, but for heavy or cumbersome articles the speed should be arranged to facilitate easy and assured loading and unloading.

For light materials, such as grain, the modern tendency is to operate the belts at speeds up to 800 feet per minute and in many of our large grain ports belts are now successfully operated at these high speeds. With narrow belts some difficulty has been met with in the loading of the belt to its correct working capacity. Wide belts having a wider feed chute permit of a good flow of the grain, but with narrow belts the width of the feed shute is consequently much less and a rapid flow cannot be obtained. From this it will be seen that narrow belts are not so applicable to these high speeds and should the feed be made more than half the width of the belt, spillage is apt to take place.

The speed of the belt is, as previously stated, influenced by the class of material being handled. With light materials, such as chaff, bran, flour, etc., the air resistance at high speeds would result in the material being blown off, while for heavy materials the discharge at the end or over trippers would cause excessive breakage in the case of anthracite coal, coke, briquettes; and with such materials as ores, the wear and tear on the belt from loading at such high speeds and the banging and hammering of the pieces on the discharge shoots would prove a considerable nuisance and cause much damage.

WIDTH OF BELTS.

The width of the belt should be determined to suit peak capacity, belt speed and size of lumps being handled.

The width of a specific capacity and speed can be found from the Capacity Table No. 23. In no instance, should the width be less than that recommended for the size of lumps as given in the table. Several combinations of width and speed may be picked. Ordinarily, the fastest speed and narrowest belt will give the greatest economy. However, when handling fragile, abrasive or lumpy material, or where good loading conditions are not possible, wider belts should be used, operated at speeds slow enough to permit the belt to be fully loaded. This decreases belt wear and breakage of materials more than enough to offset the extra cost of wider conveyors.

Package Conveyor Belt Widths.

For belts handling packages, etc., the width is determined by the size of the article and also the weight of same. Belts not less than 4" wider than the greatest outside diagonal dimension of the largest box or package, or 4" wider than the largest package in its normal position in which it is sure to be carried, should be installed. This width of belt is essential to prevent undue stretching or deflection of the belt edges, which would be the case if the packages being handled be placed on one side of the belt, or if same are as wide or wider than the belt.

A good wide belt, particularly when handling heavy or moderately heavy articles, will give better results in service than if a narrow belt is installed.

The method of feeding or the manner in which the goods are sure to be placed on the belt will also have a direct bearing on the width. If it is assured that the packages will all be of practically the same size and weight and that the attendants will place them on the belt in a regular or straight manner as shewn in Fig. 34, the belt width can be decreased accordingly. If, however, the goods are of various shapes and sizes and of different weights, the same are placed on the belt haphazard, as shewn in Fig. 35, the width of the belt will be as given previously. The width of the belt can, therefore, be considerably reduced if loading is obtained, as shewn in Fig. 34.

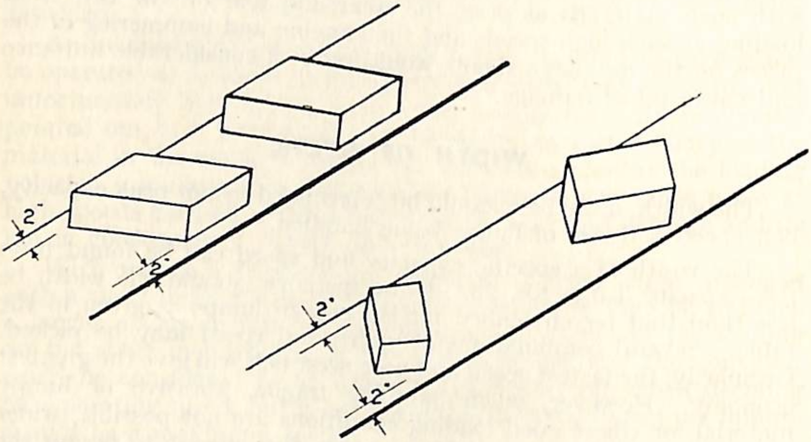


Fig. 34.

Fig. 35.

Maximum and Minimum Plies for Troughed Belts.

The belt width, as previously given, is also subject to another consideration, that is, the thickness of the belt or the number of plies required to give the correct rigidity, both longitudinally and cross-wise. The ply of the belt, as found to suit the belt tension may result in the belt being either too rigid or stiff to conform to the contour of the troughing idlers or too flexible to support the material between the spacings as given for the carrying idlers, thus producing excessive sagging and at the same time adding to the horse-power required for driving the conveyor.

Thick belts prove very troublesome, due to riding the outer inclined sides of the troughing pulleys, or running out of line over the edges of the pulleys, thus losing the steering or guiding action of the central horizontal pulleys and causing injury to the belt. Many cases of belts not running true can be traced to this cause.

Thin or flexible belts will crease lengthwise in the bends or spaces between the pulleys, and if side guide idlers are used the belt will turn back instead of receiving the guide action intended.

It is essential, therefore, to ensure the best working results and to protect the belt, that belts of correct ply should be installed. To obtain the right degree of lateral flexibility for proper troughing and the right degree of longitudinal stiffness, the number of plies should come within the range given in Table No. 28.

TABLE No. 28.

MAXIMUM AND MINIMUM PLIES FOR TROUGHED BELTS.

	Minimum Plies to Support Load.												Maximum Plies for Troughing.			
	Light Materials as Grain or Wood Chips.		Fine Coal Sand or Crushed Stone.			Fine Ores, Lump Coal, Large Stone or Gravel.				Coarse Ores or other Heavy Material.						
Belt width inches.	28 oz	32 oz	28 oz	32 oz	36 oz	28 oz	32 oz	36 oz	42 oz	32 oz	36 oz	42 oz	28 oz	32 oz	36 oz	42 oz
12	3	3	4	4	—	4	—	—	—	—	—	—	4	4	—	—
14	3	3	4	4	—	5	4	—	—	—	—	—	5	4	—	—
16	3	3	4	4	—	5	4	—	—	—	—	—	5	4	—	—
18	4	4	5	4	4	6	5	—	—	5	—	—	6	5	4	—
20	4	4	5	4	4	6	5	—	—	5	—	—	6	5	5	—
24	4	4	5	4	4	6	6	5	5	6	6	—	7	6	6	6
30	4	4	5	5	5	6	6	5	5	7	6	6	9	8	7	7
36	4	4	5	5	5	7	6	6	5	8	7	6	10	9	8	8
42	4	4	6	5	5	7	7	6	6	8	7	6	11	10	9	9
48	4	4	6	5	5	8	7	7	6	9	8	7	13	12	11	11
54	—	—	7	6	6	9	8	7	6	10	9	8	14	13	12	12
60	—	—	7	6	6	9	8	7	6	10	9	8	15	14	13	13

For flat belts and three-pulley carriers of less than 20″ troughing, the maximum number of plies, as given in Table No. 28, does not enter into account, due to the absence of lateral flexibility for troughing. The minimum number of plies as given does apply for flat belts, as this gives the correct longitudinal rigidity and should in no case be less than that specified. See also the number of plies, as determined by the tension in the belt, Table No. 4.

HORSE-POWER TO DRIVE.

The horse-power to drive belt conveyors, as compared with other forms of conveyors, is considerably less, due to the separation of the material from the moving parts, thereby reducing friction losses to a minimum.

The latest American formulas by The Jeffrey Manufacturing Co., Ltd., for determining the "effective" horse-power transmitted to the belt by the drive pulley consists of three main factors.

(1) The movement of the conveyor parts.

(2) The horizontal movement of the material to be handled.

(3) The lifting or lowering of the material.

The power required to operate factors (1) and (2) is highly variable and speculative and must provide for the following :—

(*a*) Friction losses in the idler bearings which will depend upon the character and cleanliness of the lubricant, temperature and type of bearings.

(*b*) Work done in changing the shape of the belt as it follows its catenary path over the idlers. This varies according to the stiffness of the belt, spacing of idlers and the direct tension in the belt, which varies the deflection between the idlers. Although little is known of the extent of this factor alone, tests have clearly shown that any increase in idler spacing (all other conditions being unaltered), produces an appreciable increase in the total power required.

(*c*) Work done in changing the shape of the body of the material as it passes over the idlers. This will vary with the spacing of the idlers, the tension in the belt and the nature of the material. A belt conveyor handling coke will consume more power than the same belt handling the same tonnage of clean coal. This is presumably because of the relatively sharp interlocking character of coke particles.

(*d*) The power required to elevate material through any vertical distance is frequently called the lift horse-power and involves no speculation as to its amount. When material is lowered through any vertical distance an equal amount of power is given up. Basically, declined conveyors can be figured in the same manner as inclined conveyors. As previously explained, the final horse-power is an algebraic sum in which the power required to lower material is negative. If the power to retard the material is more than the sum of the power to move the empty belt and the power to move the load horizontally, we have a retarding conveyor. When this happens the coefficient of friction which was considered conservatively high when elevating may seriously affect the resulting horse-power when the negative horse-power for decline is figured numerically larger. For this reason a correction factor of 50% of the sum of the power to move the empty conveyor and power to move the live load horizontally is added algebraically to the horse-power figure for lowering the live load. See Table No. 33 and basic horse-power formula. In practice all these power requirements are embodied in an empirical friction coefficient by which the total weight of conveyor moving parts and material is multiplied. Experience has shown that for average materials and conditions a coefficient of ·03 provides a reasonable power allowance. As the weight of the material usually exceeds the weight of the conveyor moving parts and tends to vary more in its character, any thought of using some other value for this friction coefficient should carefully entertain all variables involved. It is a fallacy to think that the factor can be changed much because of improved maintenance and supervision only, as is sometimes suggested.

The following Table No. 29 gives the values of "Q" (weights of moving parts of idlers and belt per foot centres for average conditions).

TABLE No. 29.

Width of Belt	14	16	18	20	24	30	36	42	48	54	60
Q	16·7	16·7	20	23·3	26·7	33·3	40	50	60	70	80
·03Q	·5	·5	·6	·7	·8	1·0	1·2	1·5	1·8	2·1	2·4

It will be found that this value "Q" will divide as follows :—

$\frac{2}{3}$ for carrying run and $\frac{1}{3}$ for return run.

For special conditions, or where most exact calculations are desired, Factor "Q" should be computed from the actual weight of the belt used (see pages 14 and 15), and the weight of the moving parts of the idlers given in the following Table No. 30.

TABLE No. 30.

WEIGHT OF ROTATING PARTS FOR IDLERS.

JEFFREY MANUFACTURING CO.

Width of Belt Inches	4″ Diameter Junior		5″ Diameter Reliance		6″ Diameter Reliance		6″ Diameter Hercules		6″ Diameter Defiance	
	Troughing	Return	Troughing	Return	Troughing	Return	Troughing	Return	Troughing	Return
14	12·2	10·4	17·7	14·2	24·0	18·5	—	—	—	—
16	13·3	11·5	19·1	15·6	25·8	20·3	—	—	—	—
18	14·3	12·6	20·4	17·0	27·3	22·1	—	—	45·9	44·7
20	15·5	13·8	21·8	18·4	29·1	23·9	—	—	48·3	48·3
24	17·7	16·0	24·6	21·2	32·7	27·5	—	—	53·1	55·6
30	22·1	19·5	28·8	25·4	38·1	32·8	—	—	60·1	66·5
36	—	—	34·0	29·6	44·8	38·3	64·2	43·2	69·0	77·5
42	—	—	34·0	29·6	50·2	43·6	69·4	49·9	76·1	88·4
48	—	—	42·5	38·0	55·5	49·0	74·7	56·4	83·3	99·2
54	—	—	46·7	42·2	60·9	54·4	79·9	62·9	—	—
60	—	—	50·9	46·4	66·3	59·8	85·2	69·4	—	—

Junior type idlers are of a light type.

Reliance 5″ idlers are of a good average to base upon. Reliance 6″ hercules and 6″ defiance are heavier types, respectively.

The foregoing weights must be divided by the idler spacing in feet to get the proper weight to include in value "Q."

The following Table No. 31 gives the horse-power required to drive the empty conveyor at 100 feet per minute belt speed.

For belts other than 100 feet per minute use direct proportion or the following formula :—

$$\text{H.P.} = \frac{F\ (1{\cdot}07\,L + 50)\ ({\cdot}03QS)}{1000}$$

F = Coefficient of friction for anti-friction idlers = ·03.
L = Length of conveyor in feet.
Q = Factor, see previous Table No. 29.
S = Speed of conveyor in feet per minute.

Example.—To find horse-power of an empty 24″ wide belt conveyor (anti-friction idlers) at 300 feet per minute, 200 feet centres :—

$$F = {\cdot}03\,;\quad L = 200\,;\quad {\cdot}03Q = 8\,;\quad S = 300.$$

$$\text{H.P.} = \frac{{\cdot}03\ (214 + 50)\ ({\cdot}8 \times 300)}{1000} = 1{\cdot}9\ \text{H.P.}$$

Table No. 32 gives the horse-power required to move 100 tons per hour at any belt speed.

For capacities other than 100 tons per hour, use direct proportion or the following formula :—

$$\text{H.P.} = \frac{F\ (1{\cdot}07L\ +\ (50)\ (1\tfrac{1}{8} \times C)}{1000}$$

F = Coefficient of friction for anti-friction idlers = ·03.
C = Capacity in tons per hour.
L = Length of conveyor in feet.

Example.—To find horse-power to conveyor 350 tons per hour on a conveyor 200 feet centres, using anti-friction idlers.

$$F = {\cdot}03\,;\quad C = 350\,;\quad L = 200$$

$$\text{H.P.} = \frac{{\cdot}03\ (214 + 50)\ (1\tfrac{1}{8} \times 350)}{1000}\quad 3{\cdot}1\ \text{H.P.}$$

Table No. 33 gives the horse-power required to elevate 100 tons per hour of material at any speed.

TABLE No. 31.

H.P. TO DRIVE EMPTY CONVEYORS AT 100 FT. PER MIN. BELT SPEED.

Belt width inches.	H.P. for Various Conveyor Centres in Feet.																					H.P. for each 10 ft. centres.
	50	75	100	150	200	250	300	350	400	450	500	550	600	650	700	750	800	850	900	950	1000	
14	·162	·202	·243	·324	·406	·486	·567	·648	·730	·811	·891	·937	1·05	1·15	1·22	1·31	1·38	1·47	1·54	1·63	1·70	·016
16	·162	·202	·243	·324	·406	·486	·567	·648	·730	·811	·891	·937	1·05	1·15	1·22	1·31	1·38	1·47	1·54	1·63	1·70	·016
18	·195	·244	·293	·390	·487	·585	·682	·780	·877	·973	1·07	1·17	1·26	1·36	1·46	1·56	1·66	1·76	1·85	1·95	2·04	·019
20	·227	·283	·340	·454	·567	·680	·795	·980	1·02	1·14	1·25	1·36	1·48	1·58	1·70	1·82	1·93	2·04	2·16	2·27	2·39	·023
24	·260	·325	·390	·520	·651	·780	·910	1·04	1·19	1·31	1·43	1·56	1·69	1·82	1·95	2·08	2·22	2·34	2·47	2·60	2·73	·026
30	·324	·405	·486	·650	·810	·973	1·13	1·30	1·47	1·63	1·79	1·95	2·11	2·28	2·44	2·60	2·76	2·92	3·08	3·24	3·40	·232
36	·390	·487	·585	·780	·975	1·17	1·36	1·56	1·75	1·95	2·14	2·33	2·54	2·73	2·92	3·12	3·31	3·51	3·70	3·90	4·08	·039
42	·486	·608	·730	·975	1·22	1·46	1·70	1·93	2·20	2·43	2·63	2·92	3·16	3·40	3·65	3·90	4·16	4·37	4·62	4·85	5·12	·048
48	·585	·730	·875	1·17	1·47	1·75	2·13	2·35	2·64	2·92	3·21	3·51	3·80	4·09	4·38	4·66	4·97	5·26	5·35	5·85	6·11	·059
54	·682	·850	1·02	1·36	1·70	2·04	2·39	2·73	3·06	3·40	3·75	4·09	3·43	4·79	5·11	5·45	5·79	6·14	6·46	6·81	7·16	·069
60	·778	·971	1·17	1·51	1·95	2·34	2·73	3·11	3·51	3·90	4·28	4·67	5·06	5·45	5·83	6·24	6·61	7·00	7·40	7·78	8·16	·078

To allow for terminal friction, assume H.P.s for all conveyors below 50′-0″ centres equivalent to H.P. for 50′-0″ centres.

TABLE No. 32.

H.P. TO CONVEY 100 TONS OF MATERIAL PER HOUR AT ANY SPEED.

Width Belt in inches.	H.P. For Various Conveyor Centres In Feet.																				H.P. for each 10 ft. centre.
	* 50	75	100	150	200	250	300	350	400	450	500	550	600	650	700	750	800	850	900	950	1000	
All widths. T	·41	·455	·74	1·09	1·47	1·83	2·2	2·55	2·92	3·29	3·65	·036	·547	·91	1·28	1·65	2·01	2·37	2·75	3·6	3·46	3·83

* In order to allow for terminal friction horse-powers for all conveyors below 50′-0″ centres equivalent to horse-power at 50′-0″ centres.

TABLE No. 33.

H.P. TO ELEVATE 100 TONS PER HOUR AT ANY SPEED.

Lift in feet.	5	10	15	20	25	30	35	40	45	50	55	60	70	80	90	100
Horse-power.	·56	1·68	2·18	3·93	5·06	6·18	7·87	10·12	1·12	2·25	3·37	4·5	5·62	6·75	9·0	11·25

For capacities other than 100 tons per hour use direct proportion or use the following table :—

$$\text{H.P.} = \frac{1\frac{1}{8}\,C \times H}{1000} = \frac{C \times H}{890} \text{ approx.}$$

C = Capacity in tons per hour.

H = Vertical lift in feet which material is raised. For lowering or decline conveyors the H.P. is regenerative and should be deducted.

TABLE No. 34.

H.P. FOR TRIPPERS.

Factor.		Width of Belt—Inches.						
		14-16	18-20	24	30	36	42	48
X	Plain Bearing Tripper	·0033	·0035	·0066	·0070	·0120	·0130	·0173
	Anti-Friction Bearing Tripper.	·0020	·0020	·0038	·0048	·0071	·0075	·0100
Y		·0037	·0037	·0046	·0055	·0055	·0055	·0063

Table No. 34 gives the additional horse-power required for a tripper or throw off carriage which can be obtained from the following formula :—

H.P. of Tripper = $(SX + CY) \times 1{\cdot}12$.

S = Conveyor belt speed in feet per minute.

X = Factor (Table No. 34).

C = Capacity tons per hour.

Y = Factor (Table No. 34).

Summarising the foregoing we can obtain the basic H.P. required at drive pulley or belt as follows :—

Horizontal loaded conveyors :—

$$\text{H.P.} = \frac{F\ (1{\cdot}07L + 50)\ (1\tfrac{1}{8}C + {\cdot}03QS)}{1000}$$

Inclined loaded conveyors :—

$$\text{H.P.} = \frac{F\ (1{\cdot}07L + 50)\ (1\tfrac{1}{8}C + {\cdot}03QS)\ +\ (1\tfrac{1}{8}CH)}{1000}$$

F = ·03 coefficient of friction for anti-friction idlers.
L = Length of conveyor in feet.
C = Capacity in tons per hour.
Q = Factor from table.
S = Speed of conveyor in ft./min.
H = Vertical lift in feet.

For decline conveyors (straight runs) it is necessary to calculate the three following formulae separately and use the largest numerical answer for the required H.P.

$$(1)\qquad \text{H.P.} = \frac{F\,(1{\cdot}07L+50)\ (1\tfrac{1}{8}C+{\cdot}03QS) - 1\tfrac{1}{8}CH}{1000}$$

regular formula.

$$(2)\qquad \text{H.P.} = \frac{{\cdot}5F\ (1{\cdot}07L+50)\ (1\tfrac{1}{8}C+{\cdot}03QS) - 1\tfrac{1}{8}CH}{1000}$$

using 50% "F" factor.

$$(3)\qquad \text{H.P.} = \frac{F\ (1{\cdot}07L+50)\ ({\cdot}03QS)}{1000}$$

formulae for empty horizontal.

For conveyors having declined and horizontal runs in combination, special consideration must be given to the possibility of the horizontal portions of the conveyor being loaded only and the declined portions empty and *vice-versa.*

If a tripper is used add to the basic H.P. obtained, the H.P. obtained from tripper formula, Table No. 34.

The total H.P. at the drive pulley exclusive of the losses in the drive machinery is found by adding the losses in the pulleys in the following Table No. 35 for as many of the various types of pulleys as are used, to the basic H.P. requirements found.

Any bend pulley on the carrying side of the conveyor within a distance of the head pulley, equal to ⅔ of the conveyor centres, and providing for a total bend of over 10° should be considered as a high tension bend pulley. All others are low tension bend pulleys.

Example.—In a conveyor having, in addition to head, drive and tail pulleys, a vertical take up, one low tension bend pulley and one high tension bend pulley, the losses to be added or to be subtracted from the basic H.P., are as follows :—

Vertical take up, ...	·06
High tension bend pulley, ...	·02
Low tension bend pulley, ...	·01
Total, ...	·09

TABLE No. 35.

POWER LOSSES IN CONVEYOR PULLEYS.

Type of Pulley.	Sleeve Bearing Pedestals.	Anti-Friction Bearing Pedestals.
Head, drive and tail pulleys.	·07*	·035
High tension bend pulleys.	·02	·01
Low tension bend pulleys.	·01	·005
Vertical take ups including the two bend pulleys.	·06	·03

* This value is included in the foregoing H.P. formula. Any bend pulley on the carrying side of the conveyor within a distance of the head pulley, equal to $\frac{2}{3}$ of the conveyor centres, and providing for a total bend of over 10° should be considered as a high tension bend pulley. All others are low tension bend pulleys.

Total H.P. at the headshaft = $1{\cdot}09 \times$ basic H.P. if H.P. is positive or $\cdot 91 \times$ basic H.P. if the H.P. is negative.

As will be seen from table, where anti-friction pedestals are used the losses are only one-half of those of sleeve or bushed pedestals.

Power is consumed in every pulley and its bearings on a belt conveyor and must be considered in determining the maximum H.P. requirements. Since practically all belt conveyors have a head, drive and foot pulley the losses for these are averaged at 7% and are included in the foregoing basic H.P. formula.

The power consumed in any pulley depends upon the amount of belt tension present, the angle of wrap of the belt around the pulley and the type of pedestal bearing used.

Tripper pulleys are not considered in the foregoing table but are treated as already given on page 96.

The Stephens Adamson Manf. Co. of the U.S.A. give a similar formula, based on the same principles but with slightly different factors.

They sub-divide the "effective" H.P. transmitted to the belt by the drive pulley into four main factors. The Motor H.P. can be determined by adding a fifth factor covering drive losses.

1. Power to drive the empty belt conveyor at the requred speed.
2. Power to convey the peak load over the horizontal distance

3. Power to elevate (or lower) the peak load over the nett change of elevation.
4. Power to operate trippers and other accessories.
5. Power lost in transmission from motor to head shaft.

(1) H.P. to drive empty conveyor at 100 ft/min. belt speed.
Using normal duty anti-friction idlers, plain bearings for drive and take up, Table No. 36.
Using highest grade anti-friction idlers, Table No. 37.
Anti-friction bearings for drive and take up.

The foregoing lists include the friction of the idlers, carrying belt round pulleys and the speed of operation.

(2) Power to convey the material horizontally.
Any speed, any material.
Using normal duty anti-friction idlers.
Plain bearing drive and take up terminals, Table No. 38.
Using highest grade anti-friction idlers, with anti-friction drive and take ups, Table No. 39.

The power to elevate (or lower) the peak load over the nett change in elevation is the same as in the previous formula and can be obtained from Table No. 33.

The H.P. values as given in tables have been derived from the following formula.

$$\text{H.P.} = (\text{H.P. to drive empty belt}) \ \frac{C\,(L + LO)\,(\cdot 03QS)}{1000}$$

$$+ \ \text{H.P. to convey material} \ \frac{1\tfrac{1}{8}\,C\,(L + LO)\,T}{1000}$$

$$\pm \ \text{H.P. to elevate material or lower} \ \frac{(1\tfrac{1}{8}\,TH)}{1000}$$

C = Friction factor.
= ·030 for normal duty anti-friction idlers and plain end pedestals.
= ·022 for high grade anti-friction idlers and anti-friction end.
Q = Weight of moving parts per foot, Table No. 40.
L = Horizontal centres in feet.
LO = Length constant.
150 for normal duty anti-friction idlers and plain bearings at pulleys.
200 for heavy grade anti-friction idlers and anti-friction bearings at pulleys.
S = Belt speed in feet per minute.
T = Capacity in tons per hour.
H = Vertical height in feet.

TABLE No. 36.

H.P TO DRIVE EMPTY CONVEYOR FOR EACH 100 FT./MIN. BELT SPEED.

Width in inches.	Centre to Centre Length L in Feet.													
	50	100	150	200	250	300	400	500	600	700	800	1000	1200	1400
12	·3	·3	·4	·5	·5	·6	·8	·9	1·0	1·2	—	—	—	—
14	·3	·4	·4	·5	·6	·7	·8	·9	1·1	1·2	—	—	—	—
16	·3	·4	·5	·6	·6	·7	·9	1·0	1·2	1·4	—	—	—	—
18	·3	·4	·5	·6	·7	·8	·9	1·1	1·3	1·5	1·6	2·0	—	—
20	·4	·5	·6	·7	·8	·9	1·0	1·2	1·4	1·6	1·8	2·2	—	—
24	·5	·5	·7	·8	·9	1·0	1·3	1·5	1·7	1·9	2·2	2·6	—	—
30	·6	·7	·8	1·0	1·2	1·3	1·6	1·9	2·2	2·5	2·8	3·3	3·9	—
36	·7	·9	1·0	1·3	1·5	1·6	2·0	2·4	2·7	3·1	3·5	4·2	4·9	—
42	·9	1·1	1·3	1·6	1·8	2·0	2·5	2·9	3·3	3·8	4·2	5·1	6·0	—
48	1·1	1·3	1·6	1·9	2·1	2·4	3·0	3·5	4·0	4·6	5·1	6·2	7·2	8·3
54	1·3	1·6	1·9	2·2	2·5	2·8	3·5	4·1	4·7	5·3	6·0	7·2	8·5	9·7
60	1·5	1·8	2·2	2·6	2·9	3·3	4·1	4·8	5·5	6·3	7·0	8·0	9·0	11·4

Using Normal Duty Anti-Friction Idlers and Plain Bearing Drive and Take Up Terminals.

TABLE No. 37.

H.P TO DRIVE EMPTY CONVEYOR FOR EACH 100 FT./MIN. BELT SPEED.

Width in inches.	Centre to Centre Length L in Feet.													
	200	300	400	500	600	800	1000	1300	1400	1600	1800	2000	2200	2400
12	·4	·5	·6	·7	·8	1·0	1·2	—	—	—	—	—	—	—
14	·4	·5	·6	·7	·9	1·1	1·3	—	—	—	—	—	—	—
16	·5	·6	·7	·8	·9	1·2	1·4	—	—	—	—	—	—	—
18	·5	·6	·8	·9	1·0	1·3	1·5	1·8	2·0	2·3	—	—	—	—
20	·6	·7	·8	1·0	1·1	1·4	1·7	2·0	2·2	2·5	—	—	—	—
24	·7	·8	1·0	1·2	1·3	1·7	2·0	2·3	2·7	3·0	—	—	—	—
30	·9	1·1	1·3	1·5	1·7	2·1	2·6	3·0	3·4	3·8	4·3	4·7	—	—
36	1·1	1·3	1·6	1·9	2·1	2·7	3·2	3·7	4·3	4·8	5·3	5·9	—	—
42	1·3	1·6	2·0	2·3	2·6	3·3	3·9	4·6	5·2	5·9	6·5	7·2	—	—
48	1·6	2·0	2·4	2·8	3·1	3·9	4·7	5·5	6·3	7·1	7·9	8·7	9·4	10·2
54	1·8	2·3	2·8	3·2	3·7	4·6	5·5	6·4	7·4	8·3	9·2	10·1	11·0	12·0
60	2·2	2·7	3·2	3·8	4·3	5·4	6·5	7·6	8·6	9·7	10·8	11·9	13·0	14·0

Using High Grade Anti-Friction Idlers and Anti-Friction Drive and Take Up Terminals.

TABLE No. 38.

ADDITIONAL H.P. TO CONVEY THE MATERIAL HORIZONTALLY—ANY MATERIAL AND ANY SPEED.

Tons per hour.	Centre to Centre Length L in Feet.													
	50	100	150	200	250	300	400	500	600	700	800	1000	1200	1400
50	·33	·45	·56	·56	·67	·79	·90	1·12	1·23	1·46	1·57	1·91	2·25	2·6
100	·67	·90	1·01	1·23	1·35	1·57	1·91	2·25	2·46	2·92	3·14	3·82	4·5	5·2
150	1·01	1·23	1·57	1·80	2·02	2·25	2·81	3·37	3·82	4·40	4·84	5·85	6·87	7·9
200	1·35	1·70	2·02	2·36	2·70	3·04	3·71	4·38	5·06	5·85	6·52	7·83	9·25	10·6
250	1·70	2·14	2·60	3·04	3·37	3·82	4·72	5·51	6·41	7·20	8·10	9·8	11·5	13·2
300	2·02	2·60	3·04	3·60	4·05	4·65	5·62	6·64	7·65	8·66	9·67	11·8	13·9	15·9
350	2·36	3·04	3·60	4·36	4·75	5·50	6·52	7·76	9·00	10·1	11·2	13·8	16·1	18·5
400	2·7	3·37	4·05	4·75	5·4	6·20	7·54	8·90	11·2	11·6	13·0	15·6	18·5	21·2
500	3·4	4·2	5·1	6·0	6·9	7·7	9·3	11·0	12·8	14·5	16·2	19·6	22·5	26
600	4·05	5·1	6·2	7·2	8·2	9·2	11·5	13·3	15·3	17·5	19·5	23·6	28·1	31·5
700	4·72	6·0	7·2	8·2	9·5	10·7	13·2	15·5	17·9	20·2	22·5	27	32·5	38
800	5·4	6·9	8·3	9·5	10·9	12·3	15·0	16·8	20·5	23·5	26	31·5	38	43
900	6·2	7·6	9·2	10·7	12·2	13·9	17·0	20	22·5	26	29	35	42	48
1000	6·9	8·5	10·3	12	13·6	15·3	18·8	22	26	29	33	40	46	53

Using Normal Duty Anti-Friction Idlers and Plain Bearing Drive and Take Up Terminals.

TABLE No. 39.

ADDITIONAL H.P. TO CONVEY THE MATERIAL HORIZONTALLY—ANY MATERIAL AND ANY SPEED.

Tons per hour.	Centre to Centre Length L in Feet.									
	200	300	400	500	600	800	1000	1400	1800	2200
200	2·0	2·5	3·0	3·5	4·5	5·0	6·0	8·0	10·0	12·0
400	4·0	5·0	6·0	7·0	8·0	10·0	12·0	16·0	20·0	24·0
600	6·0	7·5	9·0	10·5	12·0	15·0	18·0	23·6	30·4	36·0
800	8·0	10·0	12·0	14·0	16·0	20·0	23·6	31·5	40·5	48·4
1000	10·0	12·5	15·0	17·5	20·0	24·7	30·4	40·5	49·5	60·0
1500	15·0	18·8	22·5	26·0	30·4	37·1	45·0	60·0	75·4	90·0
2000	20·0	24·7	30·4	35·0	40·5	49·5	60·0	80·0	100·0	120·5
2500	25·0	31·5	37·1	44·0	49·5	63·0	75·4	100·0	125·0	150·0
3000	30·0	41·0	45·0	53·0	60·0	75·4	90·0	120·5	150·0	180·0

Using Highest Grade Anti-Friction Idlers and Anti-Friction Drive and Take Up Terminals.

TABLE No. 40.

AVERAGE VALUES OF Q.

Belt Width.	Q	Belt Width.	Q
12	12	30	34
14	14	36	44
16	16	42	50
18	19	48	59
20	21	54	68
24	25	60	76

Power to operate trippers and other accessories as given by S.A. Manufacturing Co. is given in the following table :—

TABLE No. 41.

ADDITION H.P. REQUIRED FOR EACH TRIPPER.

Belt width in inches.	12	14	18	20	24	30	36	42	45	54	60
H.P. to add for fixed or hand propelled tripper.	·40	·70	1·00	1·35	1·75	2·50	3·25	4·5	6·0	7·5	9·0
H.P. to add for self propelling tripper.	·44	·77	1·10	1·50	1·90	2·75	3·35	5·0	7·0	8·0	10·0

While power driven belt cleaning brushes take more power than the spring loaded type belt cleaners, the additional H.P. can be neglected except in the case of high speed brushes and some allowance should be made depending upon width of belt, speed and capacity of conveyor.

Power lost in transmission from motor to conveyor head shaft depends upon the type of drive or speed reduction gear unit used and is expressed in percentages as follows :—

TABLE No. 42.

Cast Tooth Gears.	Cut Tooth Gears, Roller drive chain and belt drive.	Spur or helical type self contained speed reducer.	Worm reduction gears.
Add 10% for each reduction.	Add 5% for each reduction.	Add 5%.	Use makers efficiency rating.

Size of Motor Required.

The total of all five factors previously given or listed, gives the H.P. required to run the conveyor and a motor of equal or greater H.P. must be used.

If the conveyor is to be started under load frequently, a high starting torque motor should be used.

BELT TENSIONS.

Determination of Belt Tensions.

Obviously to drive a belt conveyor tension must be applied by the drive pulley to pull the belt. This is called the "Effective Tension" or H.P. pull.

In addition a certain amount of mutual belt tension must be maintained to enable the pulley to grip and pull the belt. With the conveyor in operation there is a difference of belt tension on the two sides of the drive pulley. The larger tension T_1 is the maximum or tight side tension, which includes the effective tension and the mutual tension. On the other side is the mutual tension only, T_2, which is usually called the slack side tension.

Because the head drive pulley is not driven by the belt the losses in it should be omitted in determining the various tensions in the belt. This is accomplished by using a factor of ·96 in the following formula for determining the effective belt tension or H.P. pull in the belt.

Effective Tension.

This is H.P. pull which actually does the work of moving the belt, and is the difference between the tight side or total tension and the slack side tension.

$$\text{H.P. Pull or Effective Belt Tension.} = \frac{\cdot 96\ \text{H.P.} \times 33{,}000}{\text{Speed of belt in ft/mins.}}$$

in which the H.P. is the total required to operate the conveyor. If anti-friction journal bearings are used on the headshaft, the factor ·96 in the foregoing formula becomes ·98 and the ·07 factor in the Jeffry Co. formula becomes ·035.

Formulas for obtaining the effective H.P. pull for some of the usual paths of belt conveyors are given in Table 44. Formulas for the many other possible paths will be apparent by comparison with these.

Examples are given in the following pages.

TABLE No. 43.

FRICTION FACTORS.

PART OF CONVEYOR.	FRICTION FACTOR C
Revolving parts of ball or roller bearing idlers.	·02
Empty belt on ball or roller bearing idlers.	·02
Material only on belt on anti-friction idlers.	·03
Tail, bend and snub pulleys and shafts ball and roller bearings.	·01
Tail, bend and snub pulleys and sleeve bearings.	·02
Take up pull or counterweight, ball and roller bearings.	·01
Take up pull or counterweight sleeve bearings.	·02
Drive pulley assembly, ball and roller bearings.	·01
Drive pulley assembly, sleeve bearings.	·02
Fabric belt including material carried sliding on polished steel.	·25 — ·30
Friction surface belt including material carried sliding on polished steel.	·30 — ·35
Rubber surfaced belt including material carried sliding on polished steel.	·45 — ·55
Fabric surfaced belt including material carried sliding on polished steel.	·30 — ·35
Friction surfaced belt including material carried sliding on polished steel.	·35 — ·40
Rubber surfaced belt including material carried sliding on polished steel.	·40 — ·50

The friction factors in the above Table No. 43, are based on proper maintenance, lubrication and alignment.

B = Weight in lbs. of one lineal foot of belt.

C = Recommended friction factors from Table No. 43.

E = Effective H.P. pull.

F = Maximum capacity in cubic feet per hour $= \dfrac{T \times 2240}{W}$

H_1 = Vertical distance in feet material will be lifted.

H_2 = Vertical distance in feet material will be lowered.

L_1 = Length in feet of horizontal runs of the conveyor.

L_2 = Horizontal length in feet of inclined runs of ascending conveyor.

L_3 = Horizontal length in feet of inclined runs of descending conveyor.

M = Weight in lbs. of material per lineal foot of belt length.

$$= \frac{F \times W}{60{\cdot}8} = \frac{2240 \times T}{60{\cdot}8}$$

Q = Weight in lbs. of revolving parts of idlers.

S = Speed of belt in feet per minute.

T = Peak rate in tons per hour.

W = Weight in lbs. per cubic foot of material being carried.

X_1 = Pull required to move carrying run of horizontal conveyor, always an opposing force.
$= L_1\,MC + L_1\,BC + QC.$

X_2 = Pull required to move carrying run of ascending conveyor, always an opposing force.
$= L_2\,MC + L_2\,BC + QC + MH_1 + BH_1$

X_3 = Pull required to move loaded run of descending conveyor when positive is an opposing force, when negative is an assisting force.
$= L_3\,MC + L_3\,BC + QC - MH_2 - BH_2.$

X_4 = Pull required to move empty carrying run of descending conveyor when positive is an opposing force when negative is an assisting force.
$= L_3\,BC + QC - BH_2.$

Y_1 = Pull required to move return of horizontal conveyor, always an opposing force.
$= L_1\,BC + QC.$

Y_2 = Pull required to move return run of ascending conveyor belt descending. When positive is an opposing force, when negative is an assisting force.
$= L_2\,BC + QC - BH_1.$

Y_3 = Pull required to move return run of a descending conveyor belt ascending. Always an opposing force. $= L_3\ BC + QC + BH2$.

Z = Pull required because of trippers, ploughs, cleaners, skirts and other auxiliaries. For belt propelled trippers Z equals S times factor which varies from ·0013 for small light trippers weighing approximately one ton to ·0042 for large trippers weighing approximately 4 tons gross weight.
This pull is not taken into consideration when using motor propelled trippers.

■ = When angle of incline is small these values may be plus (+) but on steeper inclines, they may be minus (−) and should therefore be substracted.
If the operating H.P. is known or is computed from the tables given then (E) effective H.P. pull equals

$$= \frac{\text{H.P} \times 33{,}000}{\text{Speed in ft/min.}}$$

Example.—Find E effective H.P. pull on belt to suit the following conditions :—

Material. Unsized dry bituminous coal.
Maximum Pieces. 4″ cube.
Peak Capacity. 180 tons per hour.
Weight per cu/ft. 50 lbs.
Speed of Belt. 400 ft. per minute.

From the H.P. formula on page 99, LO is given as 120 to 150 ft. of horizontal loaded belt to be added to the foot end to the actual length of the conveyor to compensate for power requirements which do not vary with the length of the conveyor. Adding a LO value of 150 ft. to the actual path of the conveyor will give a theoretical path as shown in Figs. 36 and 37, to which letters are added to denote various points on the conveyor and to designate the various runs of the belt.

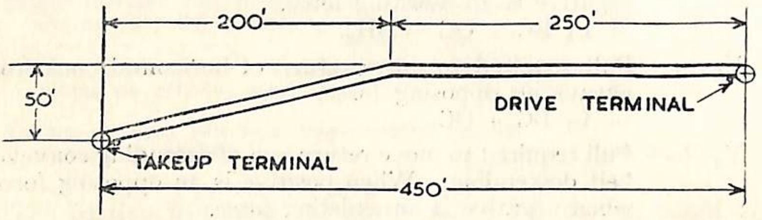


Fig. 36.

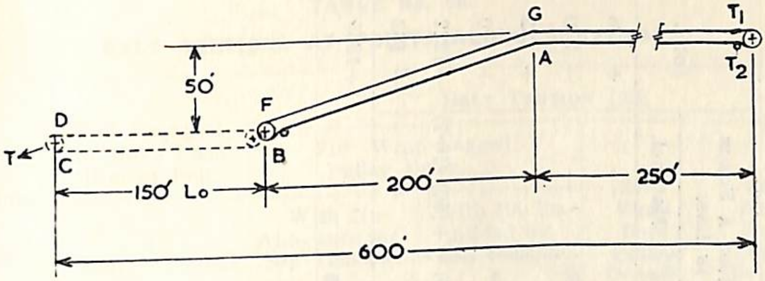


Fig. 37.

The forces accruing from the various runs or straight sections of the conveyors are shown in the tabulation, Table No. 44.

The values are computed in accordance with the formula on page 107.

B = Weight per foot of belt is assumed as 24″ wide 4·6 lbs., which is for 4-ply, 28 ounce duck belt with $\frac{1}{8}''$ × $\frac{1}{16}''$ rubber cover from Table No. 7.

Q = 33 lbs.

Normal spacing from table 5′-0″.

Belt tensions are given in Table No. 45 at various points along the conveyor as calculated for four different types of drive terminals.

The first and second columns are based on 210° wrap, lagged pulley drive having a slack side tension factor of ·38E.

The third column is based on 240° wrap, bare pulley drive having a drive factor of ·54E.

The fourth column is for 180° wrap, bare pulley drive having a drive factor of ·85E.

The first and second columns show a comparison of tensions and take up pull required between a conveyor having idlers spaced close enough to prevent undue sag and a conveyor where the belt tensions are increased for limiting the sag of the belt.

The third column shows that by the use of a 240° wrap, bare pulley drive, there will be sufficient tension at point F to prevent excessive sag and therefore no additional tension is necessary.

Note that the amount of take up tension T required increases as the tensions increase.

From the tensions thus obtained we can now select the belt.

It will be found from the tables given that a 20″ wide belt is satisfactory for the size of the largest pieces, but a 24″ wide belt is necessary to carry the tonnage within the belt speed limitations.

TABLE No. 44.

Run.	Formula.	FRICTIONAL FORCE ▲ IN LBS.			GRAVITATION A FORCE ▲ IN LBS.		Total.
		Belt.	Idlers.	Material.	Belt.	Material.	
T2A	Y_1	−(250) (4·6) (·02)	+(25) (29) (·02)	—	—	—	+ 38
A.B.	Y_2	+(200) (4·6) (·02)	+(20) (29) (·02)	—	−(50) (4·6)	—	− 210
B.C.	Y_1	+(150) (4·6) (·02)	+(14) (29) (·02)	—	—	—	+ 22
D.F.	X_1	+(150) (4·6) (·02)	+(32)■ (33) (·02)	+(150) (16·7) (·03)	—	—	+ 110
F.G.	X_2	+(200) (4·6) (·02)	+(42)▣ (33) (·02)	+(200 (16·7) (·03)	+(50) (4·6)	+(50) (16·7)	+1222
G.T.1	X_1	+(250) (4·6) (·02)	+(51)△ (33) (·02)	+(250) (16·7) (·03)	—	—	+ 183
		Effective H.P. Pull E. Algebraic Sum					+1365

▲ + designates force opposes movement of belt.
 − designates force assists movement of belt.

■ $\frac{150}{5} \times 2$ for additional under loading chute = 32.

▣ $\frac{200}{5} + 2$ for additional for convex curve = 42.

△ $\frac{250}{5} - 1$ for head pulley + 2 for convex curve = 51.

Input H.P. to belt $\frac{1365 \times 400}{33000} = 16{\cdot}5$ H.P.

TABLE No. 45.

BELT TENSIONS AT POINTS ALONG CONVEYOR.

Points.	Pull Required to Move Each Run of Belt.		Belt Tension Lbs.			
			210° Wrap Lagged Pulley Drive.			
	Run	Pounds	With No Allowance for Sag Tension ■	With 190 lbs. Added for Sag Tension ▲	240° Wrap Bare Pulley Drive⊡	180° Wrap Bare Pulley Drive△
			1	2	3	4
T_2	—	—	520	710	737	1160
A	T_2A	+ 38	558	748	775	1198
B	A B	− 210	348	538	565	988
C or D	B C	+ 22	370	560	587	1010
F	D F	+ 110	480	670	697	1120
G	F G	+1220	1720	1892	1919	2342
T_1	G T_1	+ 183	1885	2075	2102	2525
E	T_1-T_2	+1365	1365	1365	1365	1365
$T_\square$	—	—	740	1120	1174	2020

From the tabulated belt tensions, maximum tensions are as follows :—

1885 lbs. 210° wrap, lagged pulley, without addition for sag.
2075 ,, 210° ,, ,, ,, with allowance for sag.
2012 ,, 240° ,, bare ,,
2525 ,, 180° ,, ,, ,,

Taking a 24″ wide belt, 4-ply 28 oz. duck, permissible tension, 2,400 lbs.

Taking a 24″ wide belt, 4-ply 32 oz. duck, permissible tension 2,880 lbs.

The selection would be 24″ wide 28 oz. duck 4-ply because it is most economical, and suitable for the first three types of drives.

The 210° wrap lagged pulley drive with sufficient tension added to prevent excessive sag which results in a maximum belt tension. T_1 of 2075 lbs. which is 87% of the permissible belt tension of 2400 lbs.

The rubber covers will be found to be $\frac{1}{8}$″ thick on the carrying run and $\frac{1}{16}$″ thick on pulley side.

From Fig. 38, using 21·3 lbs. per foot, 4·6 for belt and 16·7 for material and idler spacing of 5 feet, a minimum tension of 670 lbs. on the carrying run is required to prevent excessive sag, subtracting 480 lbs. tension at point F, indicates that 190 lbs. should be added.

Slack Side Tension.

In order to obtain proper traction on the drive pulley or pulleys we must produce sufficient initial or slack side tension T_2 on the belt as it leaves the drive pulley. This is the initial tension and should be little more than the minimum that will enable the pulley to grip the belt. Slack side tension is accomplished by adding tension by means of an automatic counter-weighted or screw type take up. In some cases, on inclined conveyor with high lifts, the pull of the return belt produces more slack side tension than the minimum required for traction.

The counterweighted take up is preferred because it maintains a constant minimum tension on the slack side at all times, under starting loads, changes of temperature and belt stretch. When screw type take ups are used on smaller conveyors, additional or reserve belt tension should be maintained to avoid the necessity of constant adjustment. The amount of slack side tension required depends upon :—

Angle of wrap of the conveyor belt around the drive pulley or pulleys and by increasing the co-efficient of friction with pulley lagging.

By a combination of both the foregoing factors.

Table No. 46 shows drive factors "R" for varying angles of pulley wrap and for bare or lagged pulleys. (See Fig. 39).

These factors, when multiplied by H.P. pull or effective tension, give the minimum slack side tension for proper traction.

T_2 = H.P. pull or effective tension in lbs. × drive factor R. (from Table No. 46).

From Table No. 46, it can be seen that (*a*) lagging the drive pulley increases the co-efficient of friction ; (*b*) the use of snub and tandem pulleys increases the arc of contact ; (*c*) the use of a pneumatic pressure pulley as a snub with a single pulley drive makes it possible to transmit more power to the belt without increasing the slack side tension.

Tight Side Tension.

This is the maximum tension T_1 usually found at the drive pulley on the conveying side. Maximum belt tension may not always occur at the head or drive pulley but may be found at some intermediate point. Belts of irregular contours combining inclines and declines need to be analysed in detail with proper combinations

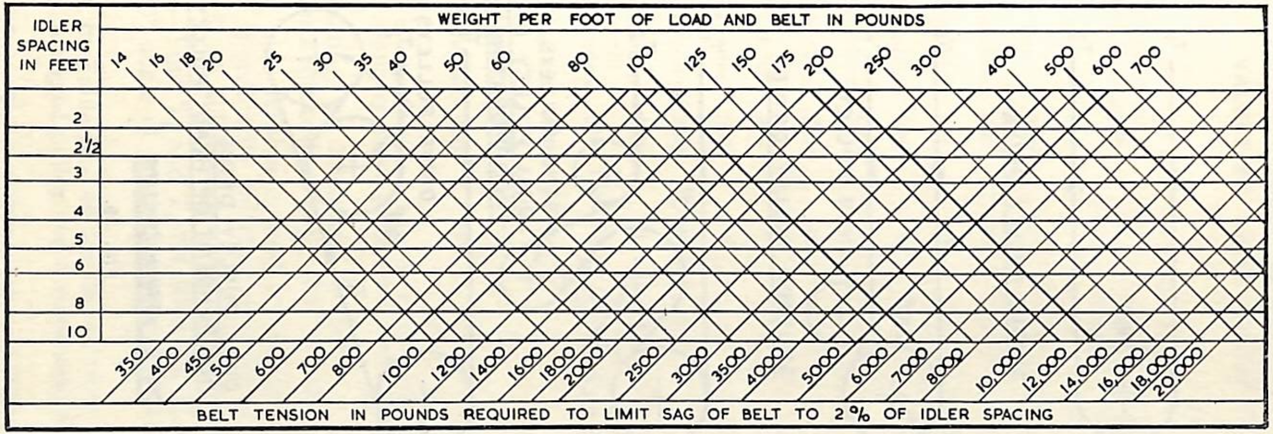
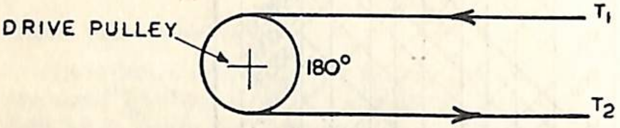
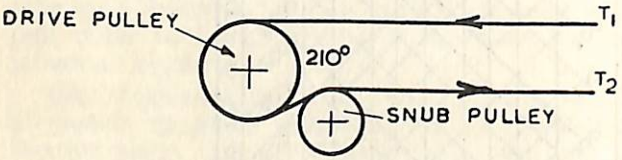

IDLER SPACING IN FEET
WEIGHT PER FOOT OF LOAD AND BELT IN POUNDS
14
16
18
20
25
30
35
40
50
60
80
100
125
150
175
200
250
300
400
500
600
700
2
2½
3
4
5
6
8
10
350
400
450
500
600
700
800
1000
1200
1400
1600
1800
2000
2500
3000
3500
4000
5000
6000
7000
8000
10,000
12,000
14,000
16,000
18,000
20,000
BELT TENSION IN POUNDS REQUIRED TO LIMIT SAG OF BELT TO 2% OF IDLER SPACING

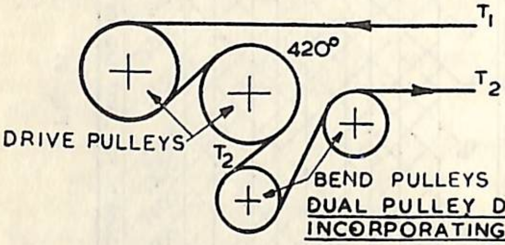
Fig. 38.



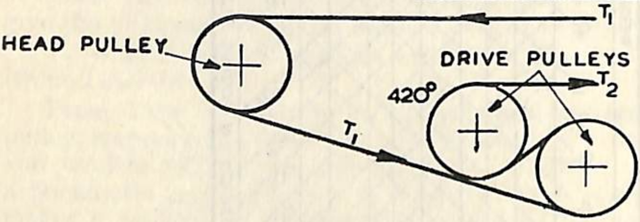
SINGLE PULLEY DRIVE



SNUBBED PULLEY DRIVE



DUAL PULLEY DRIVE
INCORPORATING HEAD PULLEY



DUAL PULLEY DRIVE
INDEPENDENT OF HEAD PULLEY

DRIVE ARRANGEMENTS

Fig. 39.

TABLE No. 46.

VALUES OF DRIVE FACTOR R.

Arc of Contact on Drive Pulleys in Degrees.			Screw Take Up.		Counterwieghted Take Up.	
			Bore Pulley.	Lagged Pulley.	Bore Pulley.	Lagged. Pulley.
Average for Single Drives.	Plain	180°	1·00	·84	·64	·50
	Snubbed	200°	·87	·72	·54	·42
	Snubbed	210°	·81	·67	·50	·38*
	Snubbed	220°	·75	·62	·46	·35
	Snubbed	230°	·70	·58	·43	·32
	Snubbed	240°	·66	·55	·40	·32
Average for Tandem Drives.	Tandem	270°	·55	·45	·32	·24
	Tandem	300°	·46	·37	·26	·20
	Tandem	330°	·39	·31	·22	·16
	Tandem	360°	·34	·26	·18	·13
	Tandem	390°	·29	·22	·15	·11
	Tandem	420°	·25	·19	·13	·09*
	Tandem	450°	·22	·16	·11	·07
	Tandem	480°	·19	·14	·09	·06

The above values marked * are most commonly encountered in ordinary single pulley and tandem pulley.

of partical loadings to determine the point of maximum tension. This is further explained on pages 105 to 112.

The tight side tension is used for selecting a belt of proper strength and is ordinarily determined as follows :—

Slack side tension and effective tension.

Example.—Find T_1 and T_2 for 3000 lbs. H.P. pull or effective tension. 270° angle of wrap, lagged pulley and counterweighted take up.

Referring to Table No. 46 for drive factor "R" corresponding to 270° wrap, a lagged pulley and counterweighted take up is found to be ·24.

Thus $T_2 = 3000 \times \cdot 24 = 720$ lbs.
Then $T_1 = 3000 + 720 = 3720$ lbs.

Table No. 47 gives in a concise form the various belt tensions for various types of tension take up, arc of contact, etc.

The top figures in each column denotes tight side tension T_1.

The bottom figures in each column denotes slack side tension T_2.

For fractions of a H.P. or greater H.P. than listed, accurate calculation may be determined by interpolation as follows :—

T_1 and T_2 for 0·2 H.P. are $\frac{1}{10}$ of the values given in the table and for 2·0 H.P. the T_1 and T_2 values are obtained by multiplying those for the 2·0 H.P. by 10.

Example.—Find the values of T_1 and T_2 for a Belt Conveyor requiring 49 H.P. running at speed of 200 F.P.M. lagged pulley with 180° wrap and using a screw take up. The table lists H.P. at 100 F.P.M., therefore 49 H.P. must be divided by 2 which gives 24·5 H.P. at 100 ft/min.

	H.P.	T_1	T_2
Multiplying values for 2 H.P. by 10 equals	20·0	12,120	5520
Reading 4 H.P. direct equals	4·0	2,420	1104
Dividing the values for 5 H.P. by 10 equals	·5	303	138
Answer	24·5	14,847	6762

The effective belt tension or H.P. pull is the difference between T_1 and T_2 or $14{,}847 - 6762 = 8085$.

The effective belt tensions or H.P. pulls at the bottom of the table are given for the various H.P. values listed in the heading for 100 foot per minute. At other speeds effective tensions or H.P. pulls are always proportional to the speed.

Example.—H.P. pull for 10 H.P. at 300 feet per minute.

$$= 33{,}000 \times \frac{100}{300} = 1100 \text{ lbs.}$$

Belt Slope Tensions.

On inclined conveyors the pull due to the weight of the empty belt on the slope may provide more slack side tension than is required for traction. The weight of the belt on the inclined runs causes tension at the top of the slope. When this occurs the corrected value of slack tension (T_{21}) should be used for obtaining the maximum belt tension T_1 This may be stated as follows :—

$$T_1 = \text{H.P. pull} + T_2 \text{ (or } T_{21}\text{)}$$

The greater pull of T_2 or T_{21} should be used in the above formula.

The following formula gives the approximate value of T_{21} :—

$$T_{21} = {\cdot}9 \text{ B.H.}$$

where B = Weight of belt per foot.
H = Vertical lift in feet.

TABLE No. 47.
BELT TENSIONS FOR DIFFERENT TYPES OF DRIVE.

	Arc of Belt Contact.		H.P. at 100 Feet per Minute.									
	Bare Pulley.	Lagged Pulley.	1·0	2·0	3·0	4·0	5·0	6·0	7·0	8·0	9·0	10·0
Using Screw Take Up	180		660 330	1320 660	1980 990	2640 1320	3300 1650	3960 1980	4620 2310	5280 2640	5940 2970	6600 3300
	200		617 287	1234 574	1850 860	2470 1150	3080 1430	3700 1720	4320 2010	4940 2300	5550 2580	6170 2870
		180	606 276	1212 552	1818 828	2424 1104	3030 1380	3636 1656	4242 1932	4848 3308	5454 2484	6060 2760
	210		598 268	1196 536	1784 804	2390 1070	2985 1335	3585 1600	4180 1870	4770 2130	4370 2400	5980 2680
	220		578 248	1156 496	1735 745	2315 995	2895 1245	3470 1420	4050 1740	4650 2010	5200 2230	5780 2480
		200	567 237	1135 475	1700 710	2270 950	2840 1190	3400 1420	3960 1650	4540 1900	5110 2140	5670 2370
		210	552 222	1100 440	1655 665	2210 890	2760 1110	3310 1330	3860 1550	4420 1780	4960 1990	5520 2220
Using Counterweight Take Up	180		542 212	1085 425	1625 635	2165 845	2710 1060	3250 1270	3790 1480	4330 1690	4870 1900	5420 2120
	200		508 178	1015 355	1525 535	2030 710	2540 890	3050 1070	3560 1250	4060 1420	4570 1600	5080 1780
	210	180	495 165	990 330	1485 495	1980 660	2470 820	2970 990	3470 1160	3960 1320	4450 1480	4950 1650
	220		482 152	965 305	1445 455	1925 605	2410 760	2890 910	3370 1060	3850 1210	4340 1370	4830 1520
	230	200	468 138	935 275	1405 415	1875 555	2345 695	2810 830	3280 970	3750 1110	4220 1250	4680 1380
	240	210	458 128	915 255	1375 385	1835 515	2290 640	2750 770	3210 900	3670 1030	4130 1160	4580 1280
		220	445 115	890 230	1335 345	1780 460	2225 575	2670 690	3120 810	3500 920	4000 1030	4450 1150
		230	435 105	870 210	1305 315	1740 420	2180 530	2610 630	3050 740	3490 850	3920 950	4350 1050
		240	428 98	855 195	1285 295	1715 395	2140 490	2570 590	3000 690	3430 790	3860 890	4280 980
		270	408 78	815 155	1225 235	1635 315	2045 395	2450 470	860 550	3270 630	3680 710	4080 780
		360	373 43	745 85	1120 130	1490 170	1860 210	2240 260	2610 300	2980 340	3350 380	3730 430
		420	355 25	710 50	1070 80	1425 105	1780 130	2140 160	2490 180	2850 210	3210 240	3550 250
		480	350 20	700 40	1050 60	1400 80	1750 100	2100 120	2450 140	2800 160	3150 180	3500 200
	Effective Belt Tension or H.P. Pull		330	660	990	1320	1650	1980	2310	2640	2970	3300

For any given condition there is a minimum slack side tension below which the drive will not operate.

The H.P. pull or effective tension is established as previously given.

Since the tight side tension T_1 is the sum of the H.P. pull or effective tension and the slack side tension, it is advisable to keep the latter as low as possible, especially on long conveyors with high H.P. requirements. In the case of inclined conveyors, the effect of belt slope tension on the total or light side tension can be neglected for :—

(*a*) Head end drive conveyors, with a rise of not more than 100 ft.

(*b*) Tail end drive conveyors with a rise of not over 25 ft.

(*c*) Lowering conveyors, with a drop of not over 25 ft.

For inclined conveyors with a greater rise or drop in elevation, a close approximation of the maximum or tight side tension can be obtained by increasing the sum of the effective and slack side tensions by a percentage value, taken from the following Table No. 48.

TABLE No. 48.

Rise H in Feet.	With Head Drive.	With Tail Drive.
1 – 24	0%	0%
25 – 49	0%	5%
50 – 74	0%	10%
75 – 99	0%	15%
100 – 125	5%	20%
125 – 149	8%	25%
150 – 174	12%	30%
175 – 200	15%	35%

from Goodyear Belt Handbook.

Percentage of increase to allow for belt slope tensions.

When drives are located in intermediate positions, use a percentage between those given in foregoing table. Where it is necessary to keep the slack side tension as low as possible it is advisable to use automatically counterweighted tensions which would maintain the actual slack side tension at very little more than necessary.

On belt conveyors using a screw type tension to develop slack side tension, the belt must be pulled up to a higher tension than is necessary to compensate for belt stretch, due to variations in temperature or other causes, which might result in the slack side tension dropping below the minimum necessary to drive.

By increasing the arc of belt contact on the drive pulley the slack side tension may also be reduced still further. Since T_2 = H.P. pull or effective tension * by factor R in Table No. 46 (drive factor table), it will be noted that using a combination with a low R factor will accomplish the desired result.

It may not, however, be desirable for economic reasons to go too far in reducing the slack side tension, especially on small or short conveyors.

If T_2 is greater than T_{21}, the tension at the take up pulley (if located at the foot end) on inclined conveyors can be approximated as follows :—

Tension in belt at take up pulley = $T_2 - T_{21}$

When tension is added at the take up pulley to compensate for belt sag, the same amount of tension should be added to the apparent value of the tight side tension T_1 to get a corrected value of maximum tension.

Loads on Pulleys, Shafts and Bearings.

Loads on driving pulleys are both radial and torsional loads. The radial load is the resultant of the tension in the belt as it approaches the pulley, sometimes called tight side tension T_1 and the tension in the belt as it leaves the pulley, sometimes called slack side tension T_2.

The resultant of these tensions depends upon the arc of contact of the belt on the pulley or the relative angles of the approaching belt and the leaving belt. The amount of direction of the resultant may be found graphically as shown in Fig. 40, where location and diameter of pulleys should correspond to scale with the actual or proposed installation and the values T_1 and T_2 should be indicated by and the resultant measured from a scale representing pounds. Where arc of contact is 180° resultant will be the sum of $T_1 + T_2$. The torsional load is the torsional or twisting moment in inch lbs, and is obtained by multiplying the radius in inches to the outside of the pulley by the difference in pounds between T_1 and T_2. Since the difference between T_1 and T_2 is called E, or the effective H.P. pull, the torsional moment can also be obtained by multiplying E in pounds by the radius in inches to the outside of the pulley.

Loads on bend idler pulleys are also both radial and torsional. However, since the tension in the approaching belt and the tension in the leaving belt is for all practical purposes equal, factors are given in Table No. 47, which when multiplied by the tension in the belt at the pulley will give the resultant radial load on the pulley.

TABLE No. 49.

Arc of Contact Degrees.	Radial Load Factor.	Arc of Contact Degrees.	Radial Load Factor.	Arc of Contact Degrees.	Radial Load Factor.	Arc of Contact Degrees.	Radial Load Factor.
5	'09	50	·85	120	1·73	190	1·99
10	·17	60	1·00	130	1·81	200	1·97
15	·26	70	1·17	140	1·88	210	1·93
20	·35	80	1·29	150	1·93	220	1·88
25	·43	90	1·41	160	1·97	230	1·81
30	·52	100	1·53	170	1·99	240	1·73
40	·68	110	1·64	180	2·00	250	1·64

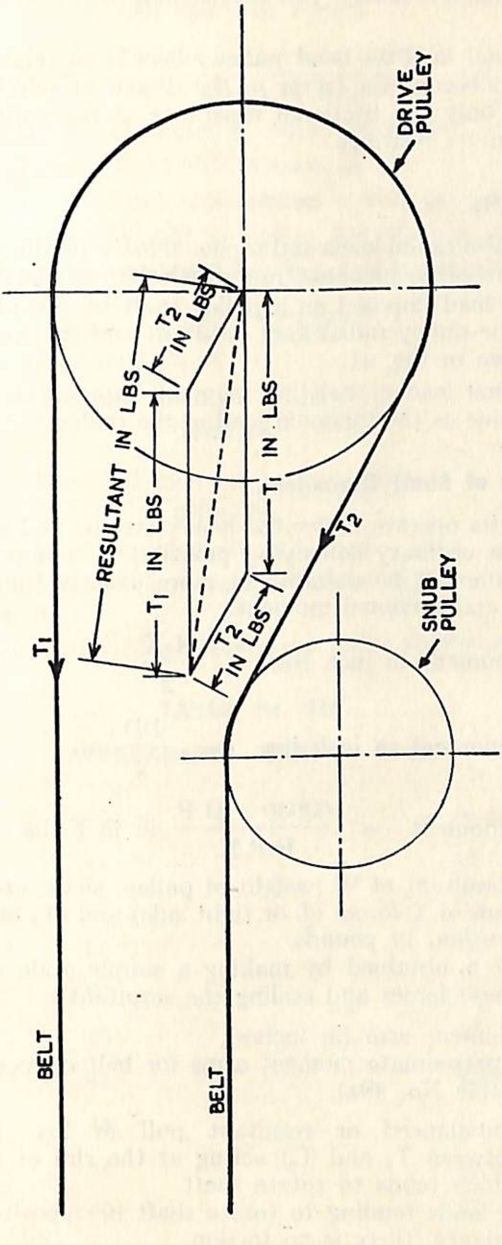

DRIVE PULLEY
T_2 IN LBS
RESULTANT IN LBS
IN LBS
T_1
IN LBS
T_1
T_2
T_2 IN LBS
T_1
SNUB PULLEY
BELT
BELT

Fig. 40.

The direction of this resultant force or load can be found by bisecting the angle between the approaching belt and the leaving belt.

The torsional load on bend pulley idlers is so relatively small that it seldom becomes a factor in the design or selection of the pulley. It is only the frictional resistance of the pulley shaft to the turning in its bearings.

Loads on Shafts.

These are also radial loads and torsional loads producing bending moments and twisting moments from which shaft can be determined.

The radial load imposed on a pulley shaft by the pulley is the resultant of the pulley radial load resultant and the weight of the pulley as shown in Fig. 41.

The torsional load or twisting moment imposed on the pulley shaft is the same as the torsional load of the pulley.

Determination of Shaft Diameters.

Where shafts operate under the heavy strains and slow speeds encountered in ordinary conveying practice, it is important that the shafts diameters be determined more exactly by computing both bending and torsional moments.

$$\text{Bending moment in inch lbs.} = \frac{R\,X}{2}$$

$$\text{Torsional moment in inch lbs.} = \frac{PD}{2} \text{ or}$$

$$\text{Torsional moment} = \frac{63030 \times \text{H.P.}}{\text{R.P.M.}} \text{ in inch lbs.}$$

R = Resultant of W (weight of pulley, shaft, etc.) and the sum of T forces (T or tight side) and (T_2 or slack side tension) in pounds.
W is obtained by making a simple scale diagram of these forces and scaling the resultant.

X = Moment arm (in inches),
Approximate moment arms for belt conveyors. (See Table No. 49A).

P = Unbalanced or resultant pull in lbs. (Difference between T_1 and T_2) acting at the rim of the pulley, which tends to rotate shaft.
If loads tending to rotate shaft in opposite direction balance, there is no torsion.

$$P = \frac{\text{H.P.} \times 33{,}000}{\text{Belt speed in ft/min.}} \text{ (in lbs.)}$$

$$P = \frac{37{\cdot}5 \times \text{T.P.H.} \times \text{feet lifted}}{\text{Belt speed in ft/min.}}$$

D = Diameter of pulley in inches.
T_1 = Tight side tension in lbs.
T_2 = Slack side tension or belt slope tension in lbs.
W = Weight of pulley and shaft in lbs.

The formula for combined bending and torsional stresses is as follows :—

$$S = \tfrac{1}{2}\,\text{S.B.} + \sqrt{ST^2 + (\tfrac{1}{2}\,SB)^2}$$

S = Combined stress.

$$SB = \frac{\text{Bending moment in inch lbs.}}{0{\cdot}098\, d3}$$

$$ST = \frac{\text{Torsional moment in inch lbs.}}{0{\cdot}196\, d3}$$

d = Diameter of shaft in inches.

For ordinary belt conveyors or feeders allow a unit stress of 12,000 lb/sq. in.

For shock loads or high speeds use value 10,000 lbs/sq. in.

TABLE No. 49A.

APPROXIMATE MOMENT ARMS.

Belt width in.	× *in inches.*
12	12
18	15
24	12
30	13
36	15
42	14 – 18
48	15 – 20
54	16 – 20
60	18 – 22

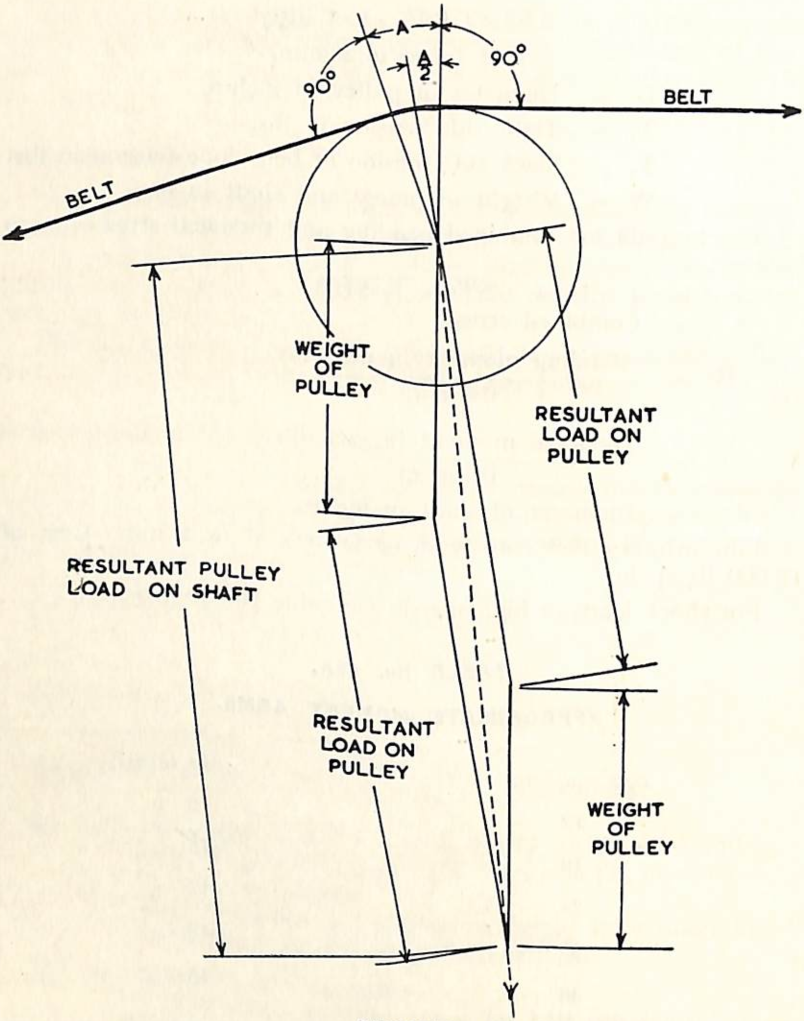

A
A/2
90°
90°
BELT
BELT
WEIGHT OF PULLEY
RESULTANT LOAD ON PULLEY
RESULTANT PULLEY LOAD ON SHAFT
RESULTANT LOAD ON PULLEY
WEIGHT OF PULLEY
DIRECTION OF RESULTANT PULLEY LOAD ON SHAFT

Fig. 41.

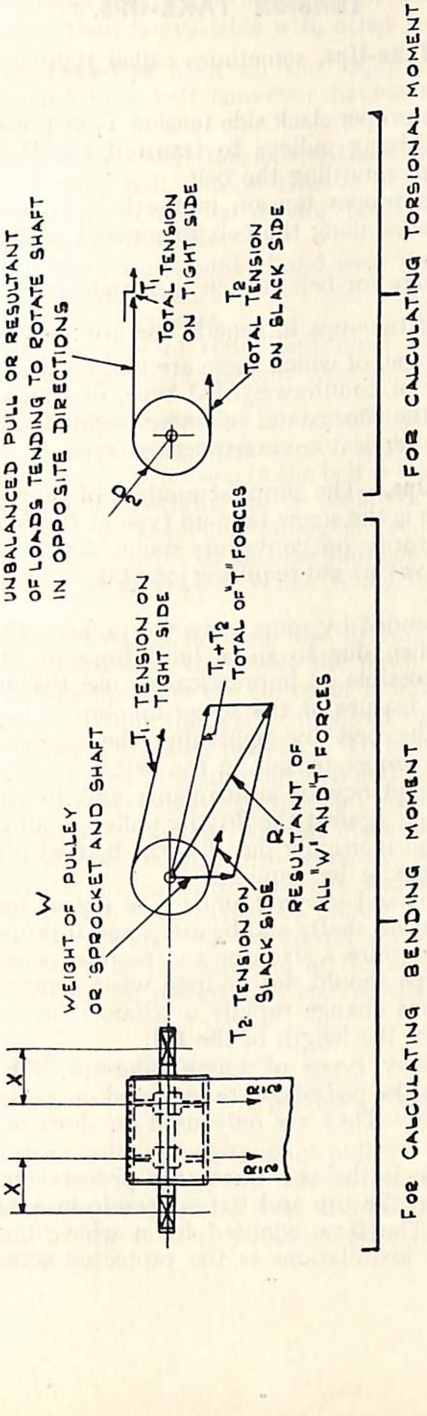

X
X
$\frac{R}{2}$
$\frac{R}{2}$
W
WEIGHT OF PULLEY OR SPROCKET AND SHAFT
T_1 TENSION ON TIGHT SIDE
T_2 TENSION ON SLACK SIDE
R
RESULTANT OF ALL "W" AND "T" FORCES
T_1+T_2 TOTAL OF "T" FORCES
P
UNBALANCED PULL OR RESULTANT OF LOADS TENDING TO ROTATE SHAFT IN OPPOSITE DIRECTIONS
$\frac{D}{2}$
T_1 TOTAL TENSION ON TIGHT SIDE
T_2 TOTAL TENSION ON SLACK SIDE
FOR CALCULATING BENDING MOMENT
FOR CALCULATING TORSIONAL MOMENT

Fig. 42.

TENSION TAKE-UPS.

The Use of Take-Ups, sometimes called tighteners or tensions, is necessary :—

- To maintain proper slack side tension T_1 or pressure of the belt on the driving pulleys to transmit the H.P. required for driving or retarding the belt.
- To maintain proper tension in the belt at loading points or other places along the belt to prevent excessive sagging of the belt.
- To compensate for belt stretch or shrinkage.

Two Types of take-ups in general use are :—

- The screw type, of which there are various designs.
- The gravity or counterweighted type, of which there are two styles, the horizontal counterweighted automatic type and the vertical counterweighted type.

Screw Take-Ups.—The simplest method of taking up belt slack in belt conveyors is the screw take-up type at the foot end, and can be used satisfactorily on conveyors under 400 ft., also for light, portable and conveyors not requiring long take-up travel or frequent adjustment.

It is recommended by some conveyor makers, that they should be used only when due to space limitations or other conditions it would be impossible or impractical to use the automatic type. The undesirable feature of the screw take-up is the possibility of over stressing the belt by tightening the screws more than is necessary to get proper tension in the belt.

To reduce frequency of adjustments and to ensure sufficient pressure of the belt against the driving pulley at all times, a greater amount of tension is usually put into the belt with screw take-ups than would otherwise be required.

This unknown and varying amount of excess tension increases the load and wear on shafts and bearings and if properly considered may sometimes require a stronger and more expensive belt.

Screw take-ups should not be used when temperature or other climatic conditions change rapidly or change sufficiently to cause wide variations in the length of the belt.

There are many types of screw take-ups, the simplest type being that where the pedestals are mounted on a slipper plate with screw adjustment. They are only used on short or light forms of belt conveyors.

A better type is the one where the pedestal or bearing block is guided both on the top and the underside in a cast-iron or fabricated frame. The type adapted for a wide range of moderate and heavy duty installations is the protected screw type. They provide ample strength with the minimum of weight and permit greater adjustment than is available with other types of take-ups.

Gravity Type Take-Ups meet all the requirements for which take-ups are needed in a belt conveyor having such advantages as :—

Maintaining constant belt tensions without adjustment.
Requiring no more belt tension than is necessary to provide proper pressure of belt on driving pulleys and to prevent excessive sag in the belt.
Minimizing power requirement of and wear on shafts, bearings and driving equipment.
Saving manual adjustments.
Reducing the number of times belt must be shortened and replaced because of permanent stretch.

Location of Tension Take-Ups is sometimes determined by convenience in supporting and access for maintenance. If the take-up is located where the tension in the belt is least, the minimum amount of pull or counterweight will be required.

When a foot end or head end pulley is used as the take-up pulley, fewer pulleys are required and the number of times the belt bends over a pulley is decreased, which tends towards increased life of the belt.

The greater the distance between the driving pulley and the take-up pulley on horizontal conveyors the more sluggish may be the effect of the take-up when the conveyor is being started, since the take-up must overcome the inertia required to start the return belt and idlers between the two pulleys. To reduce slippage and consequent belt wear, the take-up pulley should be located as near as possible to the driving pulley, so that any accumulated stretch in the belt, caused by starting, will be absorbed as quickly as possible. On the steeper inclined conveyors, the weight of a return run of belt may be sufficient to prevent starting stretch from accumulating where the belt leaves the pulley.

Screw type take-ups and the horizontal style of gravity take ups are usually placed at the foot end of the conveyor. Vertical style gravity take-ups are placed at a convenient point along the conveyor.

Amount of Pull depends upon :—

Maximum tension in belt at the position where the take-up is placed.

Angle of wrap of belt around take-up pulley.

The amount of tension in the belt can be found as given.

When the angle of wrap of the belt around the pulley is 180° the resultant pull on the take-up pulley is twice the tension of the belt at the pulley.

When the line of travel of a gravity take-up is vertical the weight of the pulley, shaft, bearings, etc., is utilized in producing pull and should be taken into consideration.

The Length of Travel of a take-up should be sufficient to compensate for the stretch of the belt, temperature and climatic conditions and provide storage for the extra length of belt which may be desired for future splices and removal of short pieces.

The amount of travel to allow for all belt stretch depends upon a great number of varying conditions and individual opinions and desires. It is usually considered that more travel should be allowed when vulcanized joints are used. 1% of the conveyor belt centres will usually be found sufficient.

Angle of Incline.

A belt conveyor which is too steep is a continual source of trouble and it may be very dangerous if the angle of incline is great enough to cause lumps of material to roll down the incline or off the belt.

Belt conveyors are ordinarily installed to carry horizontally, although most materials can be carried at an angle to the horizontal. The slope of the conveyor is governed chiefly by the class of material to be handled and the nature of the feed.

For materials which are sized the angle should be less than for large pieces mixed with fines, due to the tendency of large pieces to roll back if not carried on a layer of fines. Round materials, such as pebbles, are apt to roll back if the angle is too great and generally the limiting angle is much less than for mixed or irregular-shaped pieces.

When the feed is intermittent, such as will leave empty spaces on the belt, the material does not receive the same supporting effect as when the material is fed continuously to form one long stream on the belt, and in cases such as these the angle should be slightly less than if the latter form of feeding were arranged.

In general practice 18° or 20° is now seldom exceeded, although in the early days of belt conveyors, angles of 24° and even 26½° were adopted for belts handling coal, iron ores and similar materials. Due to the trouble which was sometimes experienced with these steep angles, the modern designer safeguards himself by limiting the angle to the figures just stated.

Table No. 50 gives what is considered to be safe angles of incline for ascending conveyors.

The angles given are conservative, but should not be exceeded. They are based on intermittent feed. The angles of incline for descending conveyors may be the same as for ascending conveyors when sluggish materials like damp earth are to be carried, but the angle should be somewhat less for materials having lower angles of repose. The combination of angle of incline and speed of the belt may not be so critical when a descending conveyor is to discharge onto a large pile or into a large bin, as there is less possibility of avalanching causing choking or flooding at the discharge. Fig. 43 shows the angle of incline and the length of the inclined portion of a conveyor when the horizontal length and the rise in feet of the inclined portion are known.

When horizontal distance is greater than 100 feet, the values can be pro-rated as shown in the following example :—

Conveyor 152 feet horizontal distance.
38 feet vertical rise.

Divide by 2 giving 76 feet and 19 feet.

The intersection of a vertical line from the 76 ft. horizontal length, with a horizontal line from a 19 ft. rise, would occur at a point corresponding approximately on the 14° line to a radius of 78½ feet, multiplying this by 2 we have an inclined length of 157 feet for the conveyor. The angle is 14° for either set of figures.

TABLE No. 50.

Material.	Angle in Degrees to Horizontal.	Material.	Angle in Degrees to Horizontal.
Beans whole	8	Ore Lumps under 4″	20
Coal, Unsized	18	Packages	15-25
,, Screwed Nuts	17	Sand, Dry free flowing	15
,, Slack	20	,, Moist	20
,, fine with lumps	20	,, Tempered Foundry	24
Coke unsized	18	Pebbles & similar rounds	14
,, fines and breeze	20	Limestone, ground clinker	18
Gravel unsized	18	Stone, crushed	18
,, sized washed	12	,, unsized 4″ over	16
,, ,, unwashed	15	,, fines ⅝″ and under	20
Grain	15/18	Wood Chips	27
Ore. Lumps 4″ and larger	18		

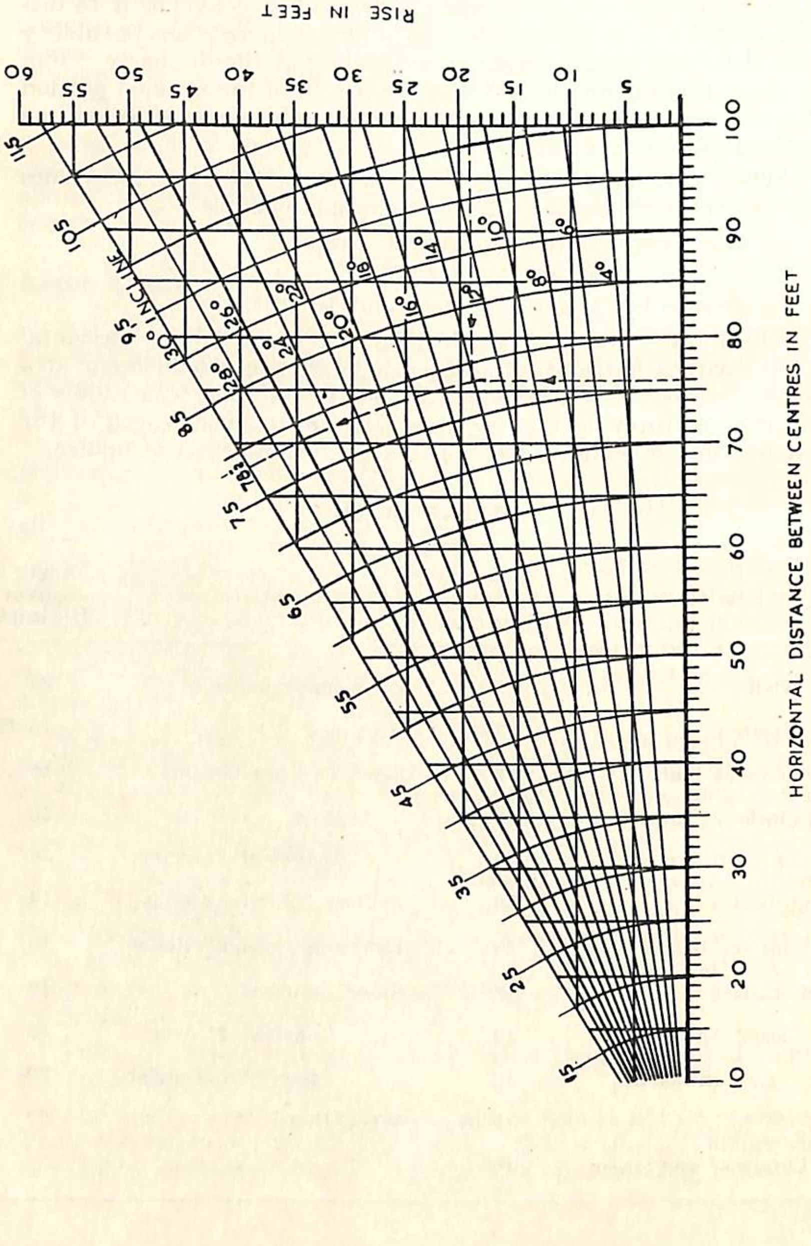

RISE IN FEET
5
10
15
20
25
30
35
40
45
50
55
60
HORIZONTAL DISTANCE BETWEEN CENTRES IN FEET
10
20
30
40
50
60
70
80
90
100
INCLINE
4°
6°
8°
10°
12°
14°
16°
18°
20°
22°
24°
26°
28°
30°
15
25
35
45
55
65
75
78½
85
95
105
115

Fig. 43.

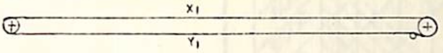
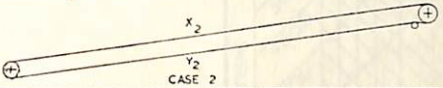
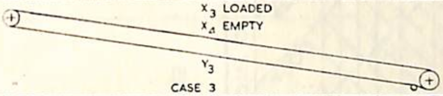
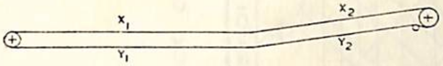
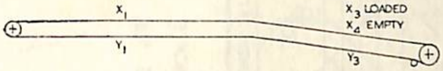
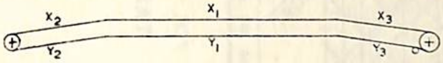
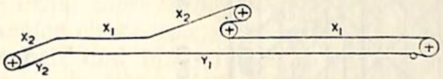
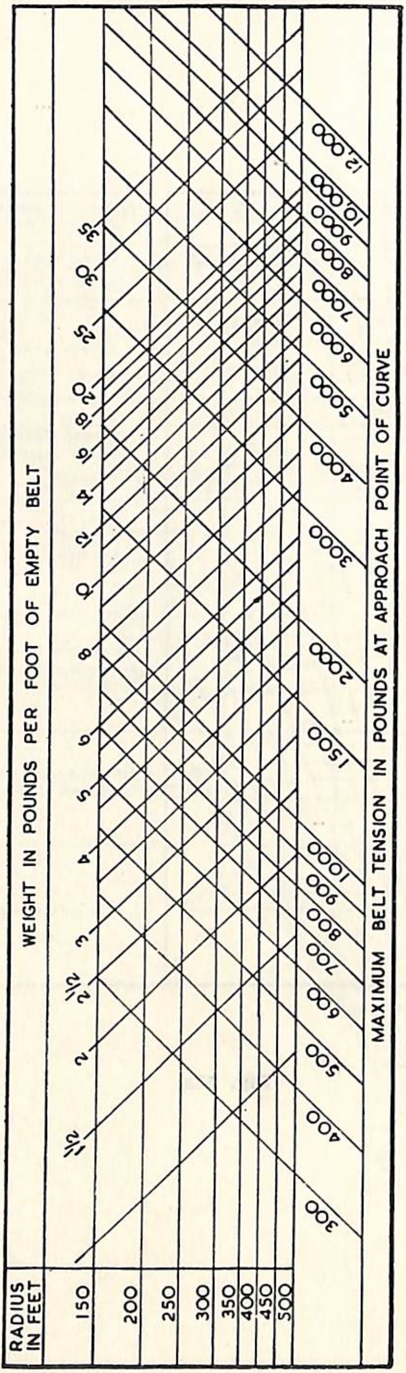
PATH OF CONVEYOR	FORMULA FOR EFFECTIVE HORSEPOWER PULL
 CASE 1	$E = X_1 + Y_1$
 CASE 2	$E = X_2 + Y_2$
 CASE 3	E, LOADED CONVEYOR = $X_3 + Y_3$ E, EMPTY CONVEYOR = $X_4 + Y_3$
 CASE 4	$E = X_1 + X_2 + Y_1 + Y_2$
 CASE 5	E, LOADED CONVEYOR = $X_1 + X_3 + Y_1 + Y_3$ E, WITH DESCENDING PORTION EMPTY AND HORIZONTAL PORTION LOADED = $X_1 + X_4 + Y_1 + Y_3$
 CASE 6	E, FOR FULLY LOADED BELT = $X_1 + X_2 + X_3 + Y_1 + Y_2 + Y_3$ INVESTIGATE FOR PARTIALLY LOADED
 CASE 7	$E = X_1 + X_2 + Y_1 + Y_2 + Z$

TABLE 43A

Fig. 43a.

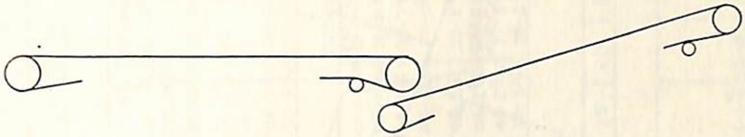


MINIMUM RECOMMENDED RADII FOR CONCAVE VERTICAL CURVES

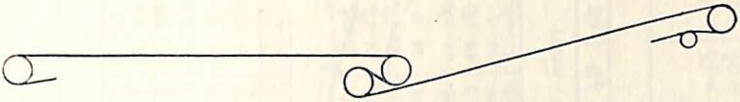
TABLE No. 51.

Radius of Curves.

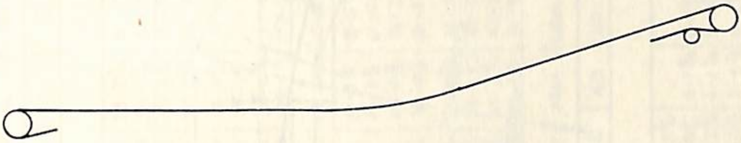
Bend pulleys or two separate conveyors may be necessary for some installations because space does not permit the proper curve.



TWO SEPARATE CONVEYORS



BEND PULLEYS



CONCAVE CURVE

Fig. 44.

They should be avoided wherever possible. In addition to requiring more bend pulleys, shafts and pedestals, together with transfer chute and skirt plates, they cause more load impacts on the belt and more frequent bending of the belt. Two separate conveyors are preferred instead of bend pulleys, even though an additional drive would be required, but the belt tension may be decreased enough so that, in some cases, a less expensive belt may be used.

The concave bend is therefore recommended whenever space will permit. (Fig. No. 44).

Table No. 51 gives the minimum recommended radii for concave vertical curves of belts gradually accelerated. Across the line starters should be avoided unless the radius is made at least 50% greater than the minimum determined from chart.

The following table gives co-ordinates for plotting concave curves.

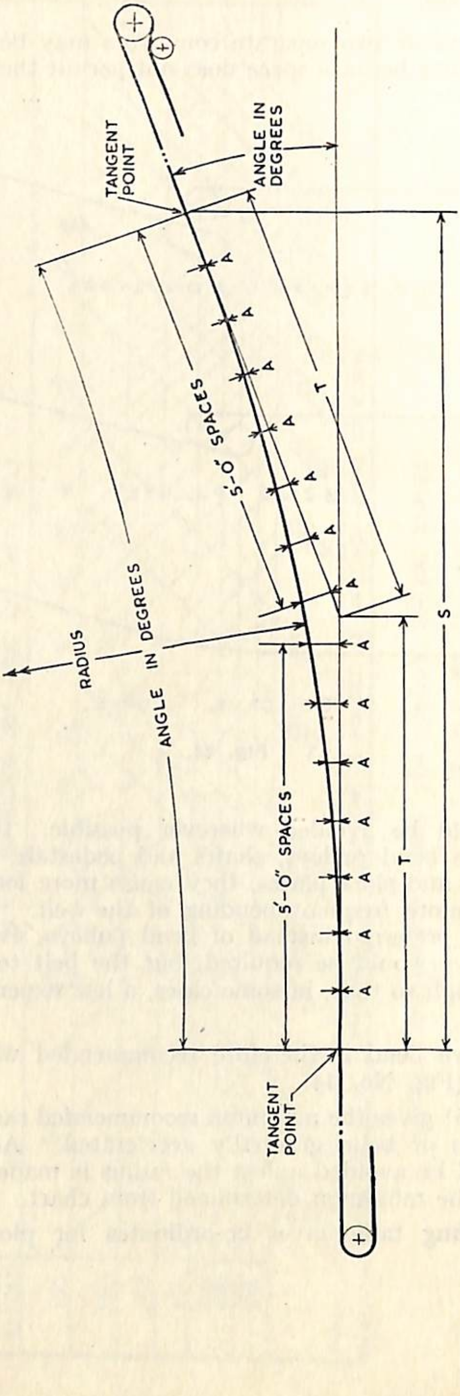

TANGENT POINT
ANGLE IN DEGREES
A
5'-0" SPACES
T
S
RADIUS
ANGLE IN DEGREES
TANGENT POINT

TABLE 52 · COORDINATES FOR PLOTTING CONCAVE CURVES

Radius of Concave Curve in Feet	Angle in Degrees													
	5	6	7	8	9	10	11	12	13	14	15	16	18	20
S · Base Lengh of Curve, Feet and Inches.														
150	13-0 7/8	15-8 1/8	18-3 3/8	20-10 1/2	23-5 9/16	26-0 9/16	28-7 7/16	31-2 1/4	33-8 15/16	36-3 1/2	38-9 7/8	41-4 1/8	46-4 1/4	51-3 5/8
200	17-5 3/16	20-10 7/8	24-4 1/2	27-10	31-3 7/16	34-8 3/4	38-1 15/16	41-7	44-11 7/8	48-4 5/8	51-9 3/16	55-1 9/16	61-9 5/8	68-4 7/8
250	21-9 1/2	26-1 9/16	30-5 5/8	34-9 1/2	39-1 5/16	43-4 15/16	47-8 7/16	51-11 3/4	56-2 7/8	60-5 3/4	64-8 1/2	68-10 15/16	77-3 1/16	85-6 1/16
300	26-1 3/4	31-4 5/16	36-6 3/4	41-9	46-11 1/8	52-1 1/8	57-2 15/16	62-4 1/2	67-5 13/16	72-6 15/16	77-7 3/4	82-8 3/16	92-8 1/2	102-7 1/4
350	30-6 1/16	36-7	42-7 7/8	48-8 1/2	54-9	60-9 5/16	66-9 3/8	72-9 1/4	78-8 13/16	84-8 1/16	90-7 1/16	96-5 11/16	108-1 7/8	119-8 1/2
400	34-10 3/8	41-9 3/4	48-9	55-8	62-6 7/8	69-5 1/2	76-3 7/8	83-2	89-11 3/4	96-9 3/16	103-6 5/16	110-3 1/16	123-7 5/16	136-9 11/16
450	39-2 11/16	47-0 7/16	54-10 1/8	62-7 1/2	70-4 3/4	78-1 11/16	85-10 3/8	93-6 11/16	101-2 3/4	108-10 3/8	116-5 5/8	124-0 7/16	139-0 11/16	153-10 15/16
500	43-6 15/16	52-3 3/16	60-11 1/4	69-7	78-2 9/16	86-9 7/8	95-4 7/8	103-11 1/2	112-5 11/16	120-11 1/2	129-4 15/16	137-9 13/16	154-6 1/8	171-0 1/8
T · Distance from Tangent point to Intersection, Feet and Inches														
150	6-6 9/16	7-10 3/8	9-2 1/16	10-5 7/8	11-9 11/16	13-1 1/2	14-5 5/16	15-9 3/16	17-1 1/16	18-5	19-9	21-1	23-9 1/16	26-5 3/8
200	8-8 3/4	10-5 3/4	12-2 3/4	13-11 13/16	15-8 7/8	17-6	19-3 1/8	21-0 1/4	22-9 7/16	24-6 11/16	26-3 15/16	28-1 5/16	31-8 1/8	35-3 3/16
250	10-11	13-1 1/4	15-3 1/2	17-5 3/4	19-8 1/8	21-10 1/2	24-0 7/8	26-3 5/16	28-5 13/16	30-8 5/16	32-10 15/16	35-1 5/8	39-7 1/8	44-1
300	13-1 3/16	15-8 11/16	18-4 3/16	20-11 3/4	23-7 5/16	26-2 13/16	28-10 5/8	31-6 3/8	34-2 1/8	36-10	39-5 15/16	42-1 15/16	47-6 3/16	52-10 3/4
350	15-3 3/8	18-4 1/8	21-4 7/8	24-5 11/16	27-6 9/16	30-7 7/16	33-8 7/16	36-9 7/16	39-10 1/2	42-11 11/16	46-0 15/16	49-2 1/4	55-5 3/16	61-8 9/16
400	17-5 9/16	20-11 9/16	24-5 9/16	27-11 11/16	31-5 3/4	34-11 15/16	38-6 3/16	42-0 1/2	45-6 7/8	49-1 3/8	52-7 15/16	56-2 9/16	63-4 1/4	70-6 3/8
450	19-7 3/4	23-7	27-6 1/4	31-5 5/8	35-5	39-4 7/16	43-3 15/16	47-3 9/16	51-3 1/4	55-3	59-2 15/16	63-2 15/16	71-3 1/4	79-4 3/16
500	21-10	26-2 1/2	30-6 15/16	34-11 9/16	39-4 3/16	43-8 15/16	48-1 3/4	52-6 5/8	56-11 9/16	61-4 11/16	65-9 15/16	70-3 1/4	79-2 1/4	88-2

Radius of Concave Curve in Feet	Distance from Tangent Point, Feet																	
	5	10	15	20	25	30	35	40	45	50	55	60	65	70	75	80	85	90
A · Length of Ordinate, Feet and Inches:																		
150	0-1	0-4	0-9	1-4 1/16	2-1 3/16	3-0 3/8	4-1 11/16	5-5 3/16										
200	0-0 3/4	0-3	0-6 3/4	1-0 1/16	1-6 13/16	2-3 1/8	3-1	4-0 1/2	5-1 9/16	6-4 3/16								
250	0-0 5/8	0-2 7/16	0-5 7/16	0-9 5/8	1-3 1/16	1-9 11/16	2-5 9/16	3-2 11/16	4-1 1/16	5-0 3/8	6-2	7-3 11/16	8-7 9/16					
300	0-0 1/2	0-2	0-4 9/16	0-8	1-0 1/2	1-6 1/16	2-0 1/2	2-8 3/16	3-4 11/16	4-2 3/8	5-1	6-0 3/4	7-1 1/2	8-3 3/8	9-6 5/8	10-10 3/8		
350	0-0 7/16	0-1 11/16	0-3 7/8	0-6 7/8	0-10 3/8	1-3 7/16	1-9 1/16	2-3 1/2	2-10 11/16	3-7 1/16	4-4 3/16	5-2 3/16	6-0 5/16	7-0 7/8	8-1 9/16	9-3 3/16	10-5 13/16	11-9 1/4
400	0-0 7/16	0-1 1/2	0-3 7/16	0-6	0-9 7/16	1-1 1/2	1-6 7/16	2-0 1/16	2-5 3/16	3-1 5/8	3-9 9/16	4-6 5/16	5-3 3/4	6-2 1/16	7-1 1/4	8-1	9-1 9/16	10-3 1/16
450	0-0 3/8	0-1 3/8	0-3	0-5 3/8	0-8 5/16	1-0 1/16	1-4 1/4	1-9 7/16	2-2 15/16	2-9 7/16	3-4 7/16	4-0 1/4	4-8 3/4	5-5 3/4	6-3 5/8	7-2 1/8	8-1 1/16	9-1 1/16
500	0-0 5/16	0-1 1/4	0-2 11/16	0-4 13/16	0-7 1/2	0-10 3/4	1-2 3/4	1-7 3/16	2-0 7/16	2-6 1/8	3-0 3/8	3-7 3/8	4-3 5/16	4-11 3/16	5-7 7/8	6-5 3/16	7-3 5/16	8-2 1/8

DISCHARGE OF MATERIAL FROM BELT.

Material may be removed or discharged from the belt conveyor by any of the following methods :—

1. By allowing the material to flow over the end of the head pulley. This is the simplest form of discharge.
2. Discharging over one or more fixed intermediate bend pulleys called fixed trippers.
3. Discharging over bend pulleys mounted on a carriage which travels along the belt. These are called movable trippers or travelling throw off carriages.
4. Scraping the material crosswise over one edge or both edges of the belt by means of one or more fixed or travelling ploughs.

Discharge Over End Pulley.

The material, due to having acquired the same speed or velocity of the belt, would continue to travel in the same direction and at the same speed, but two other forces, that is gravitation and air resistance, come into operation and the path of the material is diverted as shown in the following chart. The resistance due to the air is usually very small for ordinary materials, which are not of a very light nature and easily deflected, and is therefore neglected. The material can be discharged into a spout or chute or directly onto a pile. The location of discharge is therefore limited to one end of a non-reversible conveyor and to each end of a reversible conveyor.

Even though the normal operation of the conveyor may not be designed for end discharge, provision should be made for the discharge over the end pulley of any material which may not have been cleanly discharged by any of the following intermediate discharging means.

See trajectory of material over discharge chute, Fig. 53.

Discharge Over Fixed Trippers.

Where discharge is required at certain fixed and separate positions along the conveyor run, the fixed form of tripper can be used. The operation of this type of tripper is made automatic by arranging each of the discharge chutes, except the last one, with a two-way chute, one leg of which delivers the material to the hopper or bin underneath the conveyor and the other leg leading back to reload on the belt. When the hopper is full, a flap valve, which can be arranged in the chute, is operated to divert the flow onto the bye-pass leg or the material can be allowed to back up in the side chute and overflow into the return leg, from which it is fed back to the belt. (See Fig. 69).

The material so returned is then carried forward by the belt to the next tripper, where it is discharged until the hopper is filled, when it is again returned to the belt, and so on. As many as nine fixed trippers on one run have been arranged in this way.

The question of whether this is the best method of discharge depends entirely upon the layout of the conveyor, but modern practice tends to lean to the travelling tripper. (See Fig. 68).

Much wear is experienced due to the continual reloading of the material back to the belt, and also the repeated bends made by the belt help to shorten its life.

The cost of several of these devices, with their pulleys, shafts, etc., and chutes, is considerably more than that of one travelling tripper and it should be noted that the material is at each device elevated a few feet, thus adding to the H.P. required to operate the complete unit.

The fixed tripper, due to these many objections, is in fact largely going out of use, but in some instances, particularly where the hoppers are not continuous, their adoption is fully justifiable.

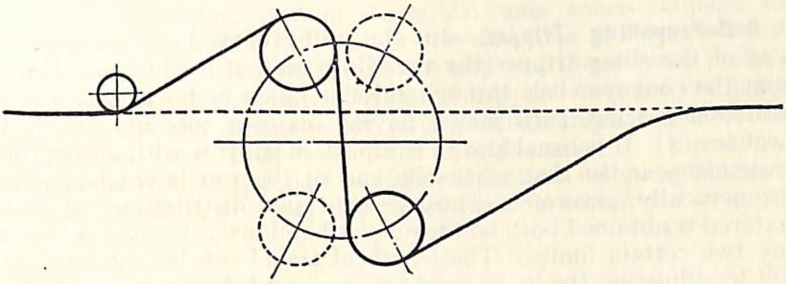


Fig. 45.

An improvement upon the fixed tripper just described is the one fitted with movable pulleys, as shown in Fig. 45. The pulleys are carried in two side frames, provided with worm wheel trunnions, placed midway, which revolve in pedestals secured to the brackets on the conveyor structure. A worm, suitably fixed, engages with these trunnions and thus the frame can be turned by hand through an angle suitable for securing a good discharge or, on the other hand, swung clear of the belt as indicated in dotted lines in Fig. 45. In this latter position the material will pass straight forward to the tripper in operation, thus obviating the lifting and reloading of the material. It is essential in a device of this kind to have gravity take-up tension gear. Another feature of this form of tripper is that should reversible conveyors be required this device answers its purpose in either direction of belt travel.

Discharge Over Movable Trippers.

Movable Trippers.—The movable tripper finds its best application for obtaining an even discharge over a row of continuous hoppers, but can also be operated to give discharge at any fixed point along the conveyor run.

Hand-Propelled Type.—The simplest form is the hand-propelled tripper, being chiefly adopted for light or heavy materials with the narrower widths of conveyor belt. These trippers are moved along from one position to another by means of a handwheel or crank handle, connection to the travelling wheels by means of a chain or spur and pinion drive. The pulleys and chute are mounted upon either a cast-iron or built-up steel frame, supported upon four travelling wheels, which operate upon suitable rails arranged on each side of the conveyor. Usually these trippers have traction on both rails and can be operated from either side.

Self-Propelling Tripper.—In the self-propelled or automatic type of travelling tripper the travelling motion is obtained direct from the conveyor belt through suitable chain or bevel and worm reduction gearing, each maker having his own specially-designed mechanism. It is usual also to equip these trippers with automatic reversing gear, so that when one end of the run is reached it is automatically reversed. Thus a constant distribution of the material is obtained both on the outward and inward travel between any two certain limits. The length of travel can be regulated at will by adjusting the location of the stops and discharge can therefore be obtained over any range without further attention after the stops are set and the conveyor started.

Motor-Driven Tripper should be used where continuous and uniform distribution of material along the conveyor is required or where the tripper is to be frequently moved to various fixed discharge locations. Being propelled by an independent electric motor, it can automatically reverse at each end of its travel by limit switches carried on the tripper and actuated by stops placed where desired along the run. Push buttons are provided for manual control to permit continued discharge at any desired location. It can be moved when the conveyor belt is moving and stationary. Electric current may be transmitted to the tripper through trolley wires or cable reeling drums carried on the tripper.

The travel speed of the motor propelled tripper is normally 30 ft. per minute.

The travel speed of the belt propelled tripper is approximately one-tenth of the belt speed of the loading belt.

Size of Pulleys.

Usually the pulleys as used on trippers are on the small side, due chiefly to the object of keeping the framing as small as possible. For tail pulleys the usual diameter in inches is four times the number of belt plies and this proportion should also apply in the case of tripper pulleys. Generally, however, smaller pulleys are instituted, which do no good to the belts, particularly heavy ones. In no case should the pulleys be made less than three inches in diameter per ply of belt. For grain belts this proportion, due to the small number of plies used, is very often adopted.

Tripper Brushes.

The adoption of a cleaning brush on any form of tripper is well-worth the small extra cost, independent of the nature of the material. Abrasive, cutting materials cause much damage to the cover of the belt if no brush is provided, due to being trapped in the reverse bending of the belt round the lower bend pulley. It is found in practice that brushes on trippers soon get made up with material and instances are known where through this cause the brushes have been taken off and discarded. If periodic attention is paid to this part of the device and a self-cleaning type of brush installed, much of this trouble will be eliminated.

Trailers.

The majority of trippers are provided with a trailer or extended framework carrying two or three sets of carrier idler sets for supporting the belt as it rises up to meet the tripper top pulley.

Tripper Chutes.

The chutes as arranged on the tripper framework can be made to give discharge to one or both sides or to give a return discharge back to the belt. By arranging a single plate flap valve, as shown in Fig. 46, either side can be fed as required, or as shown in Fig. 47, with a plate double flap valve the material can be diverted to either side or centrally as desired. Other forms of chutes are also shown.

The angle of slope of the chutes to ensure flow of material through it should be steep enough.

TABLE No. 53.

Very free flowing materials :— Grains and beans.	30 to 35°
Free flowing materials :— Clean gravel and clean sized coal.	35 to 40°
Moderately free flowing materials :— Low moisture sand, crushed stone, coal and ores.	40 to 50°
Sluggish materials, moist earth, wet coal and ores, having fines and clay.	50 to 60°

If fine material should require a steeper chute a short reversible cross belt should be used instead of a chute.

Extending Travel or Tripper.

When the tripper does not approach near enough to the loading chute for the belt in its gradual rise from the carriers to foul or be cut by the loading skirts, the conveyor and its loading chute may be installed wholly on the level, as shown in Fig. 11. This condition applies particularly to very long storage places or where room will permit of the conveyor feed point being extended beyond the hoppers. The required length of extension or the distance to allow between the tripper and the point where the belt bends from the horizontal is very variable and no fixed ruling can be given. Various dimensions are sometimes stated, but the distance to allow is best determined by the lay-out of the conveyor. In no case should the distance be cramped or stinted, as very serious damage will result to the belt if same should foul the skirts. From practical experience this distance is usually from 25 to 30 feet, measured from the centre of the tripper head pulley, for trippers of ordinary height. Should large diameter pulleys be used, necessitating a larger and higher framework, this distance will be considerably increased and provision should be allowed accordingly.

Where it is essential that the tripper should operate close up to the feed-point, the desired effect can be obtained as shown on Fig. 25.

Figs. 48, 49 and 50 illustrate trippers equipped with special features. In Fig. 48 the tripper is arranged with a four-pulley attachment, being used on reversible belt conveyors where material is delivered to the tripper from either direction.

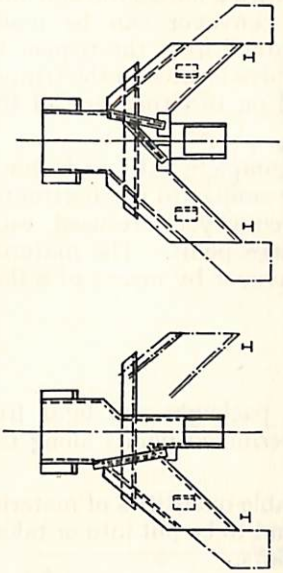


Fig. 47.

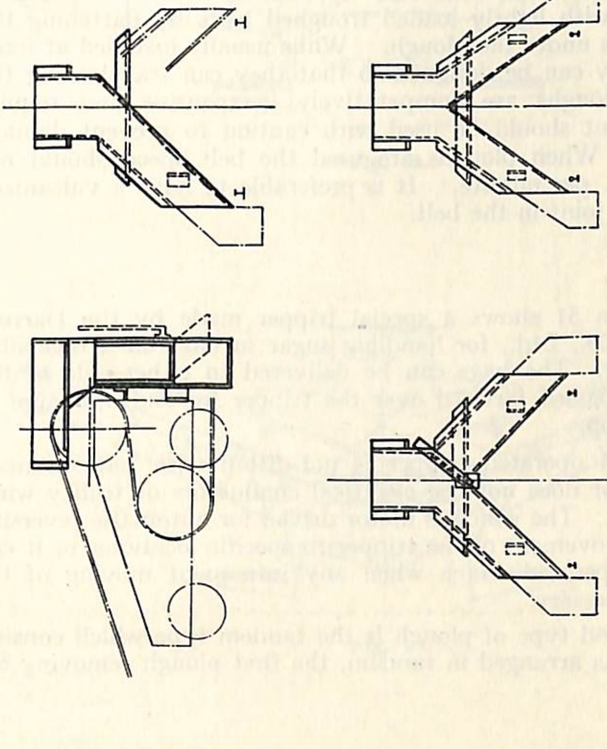


Fig. 46. ARRANGEMENTS OF THROW-OFF CARRIAGES

Fig. 49 shows a tripper fitted complete with a horizontal stacking conveyor attachment. This horizontal conveyor can be made reversible, so that material may be discharged from the tripper to either side of the conveyor. A flap gate valve located in the tripper spout may be used to direct the material on to either side of the short distributing conveyor.

Fig. 50 illustrates a tripper equipped complete with an inclined stacking conveyor, this system being very similar to the horizontal arrangement, except that two short belt conveyors are used, each being inclined to provide a higher discharge point. The material is directed to either inclined cross belt conveyor by means of a flap valve in the tripper spout.

Discharge by Ploughs or Scrapers.

These are usually used for removing packages and light free flowing granular bulk materials at predetermined points along the conveyor.

Ploughs can be made to discharge variable quantities of material from either side or both edges of the belt and to be put into or taken out of operating position from remote points.

Although they find their chief application with flat belts they can be used with lightly loaded troughed belts by flattening the troughed belts under the plough. While usually installed at fixed positions, they can be designed so that they can travel along the conveyor. Ploughs are comparatively inexpensive and require little space but should be used with caution to prevent damage to the belt. When ploughs are used the belt speed should not exceed 200 ft. per minute. It is preferable to have a vulcanized splice for the joint in the belt.

Special Types.

Illustration 51 shows a special tripper made by the Darrold Engineering Co., Ltd., for handling sugar in bags on a reversible belt conveyor. The bags can be delivered to either side of the conveyor or passed forward over the tripper for end discharge in either direction.

The winch operated tripper is not fitted with self-contained travel drive or does not use electrical conductors or trolley wires along the run. The winch is motor driven for automatic reversing or frequent movement of the tripper to specific locations, or it can be a hand operated winch when any infrequent moving of the tripper is necessary.

A very good type of plough is the tandem type which consists of two ploughs arranged in tandem, the first plough removing the

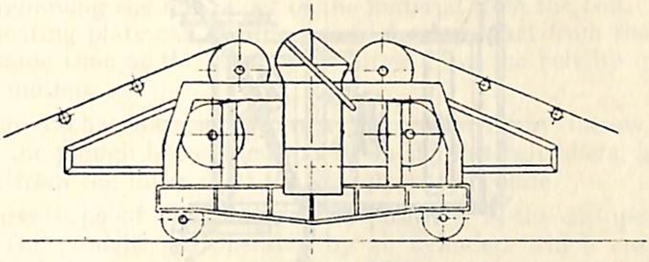


Fig. 48.

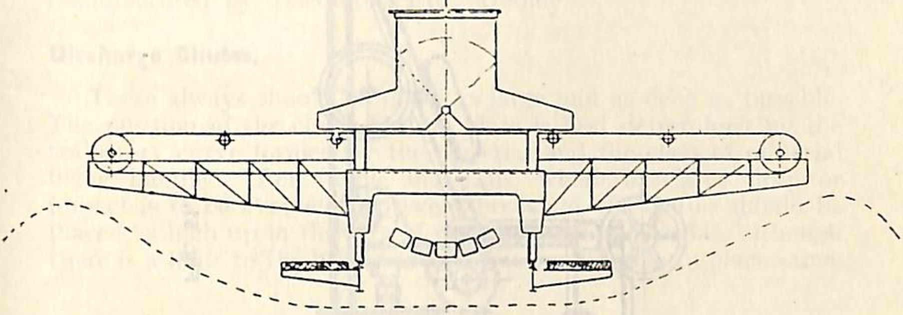


Fig. 49.

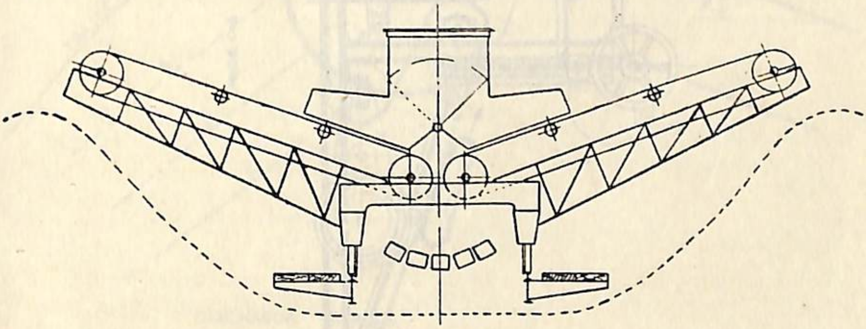


Fig. 50.

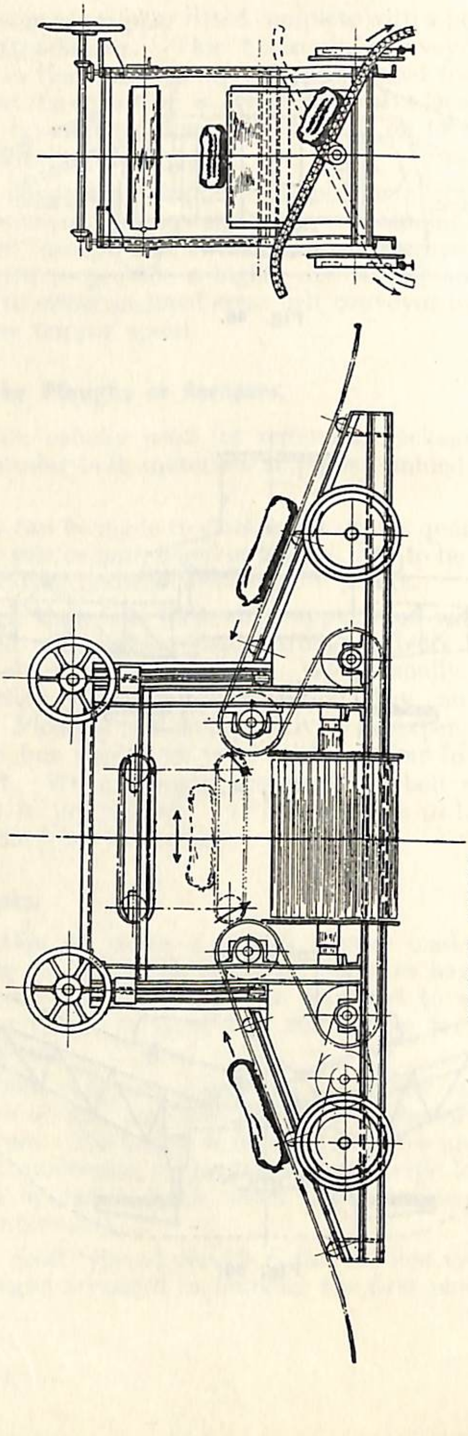


Fig. 51.

major portion of the material on the belt and the second plough finally removing the remainder of the material from the belt. The under bearing plate can also be moved from contact from the belt at the same time as the ploughs are lifted from the belt by means of link motion.

When discharging bags from a belt conveyor by means of a plough, the plough is designed to run on the flat belt idlers, lifting the belt from the idlers onto the plough bottom plate.

A later type of plough for fixed positions is the air-operated type. The ploughs are operated by air cylinders which are controlled by individual solenoid valves which are themselves electronically operated from remote control panels. The Lang Pneumatic Co., Ltd., Wolverhampton, have fitted such equipment for belt conveyors where material is required to be diverted into as many as twenty individual bins, the control panels being designed and manufactured by Teledictor, Ltd., Dudley.

Discharge Chutes.

These always should be made as large and as deep as possible. The position of the chute bottom plate is best determined by the trajectory curve formed by the material and the class of material being handled. For fragile materials, where excessive drop or impact is to be avoided to prevent breakage, the chutes should be placed as high up in the face of the discharge as possible, although there is a limit to the height at which it is advisable to place same.

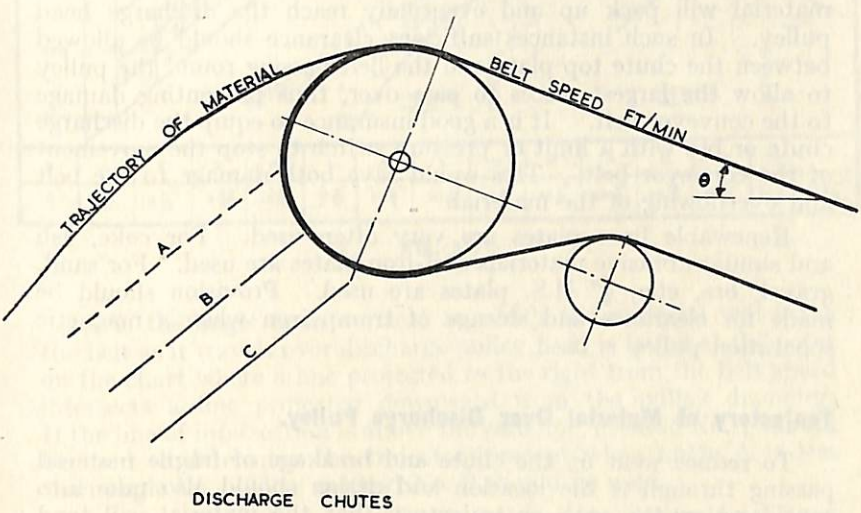


Fig. 52.

For ordinary materials which are dry and free flowing and which, therefore, do not adhere to the belt and the chute, the position of the chute can be as shown in Fig. 52, position B. The dotted position A, Fig. 52, indicates the theoretical position to prevent impact, but does not function satisfactorily in practice.

For materials which tend to stick to the belt and where no cleaning brush is installed the chute can be arranged at position C. When belt cleaners or ploughs are used the chute should be placed still further back to ensure that the brush is so arranged that the material as it is discharged from the belt clears same and that any material removed by the brush enters the chute, or in the case of ploughs low enough to catch the dribble unless separate chutes are used. See Fig. 22.

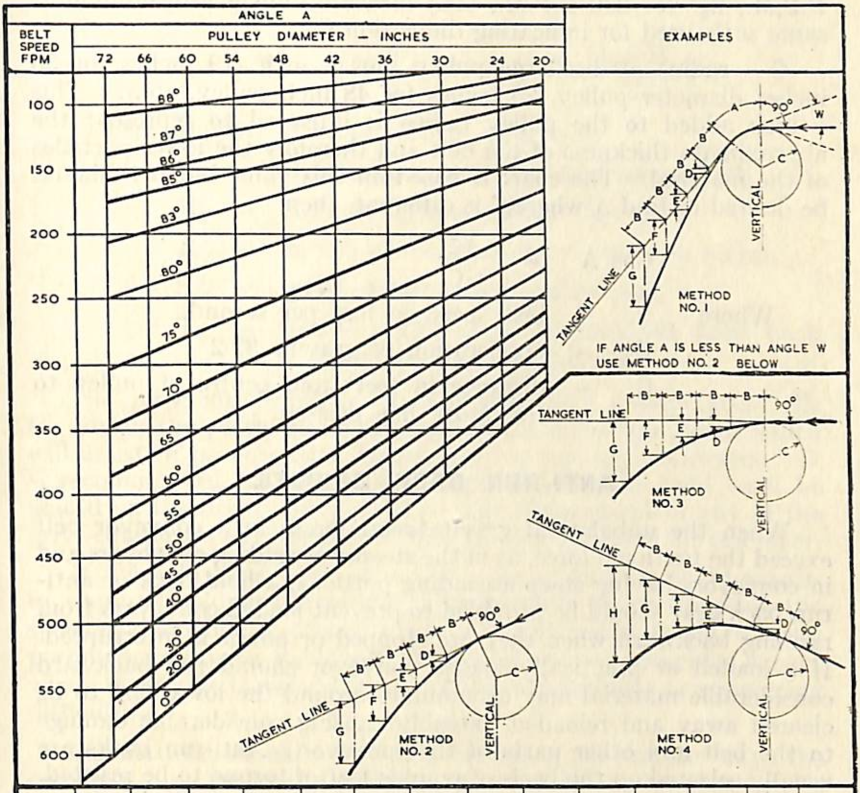
The top edge of the chute bottom plate should be arranged, with sufficient clearance from the belt passing round the pulley to ensure that pieces cannot become caught or wedged. In the lower position C this trouble is practically eliminated, but at position B material is liable to get jammed between the belt and the bottom plate edge, particularly when the conveyor is slowing down or starting up and discharging at the same time.

The thickness of chute bottom plates is best determined by the class of material being handled. In such instances as when the conveyor is delivering to a large bunker or open pile, the discharge chute can be dispensed with, allowing the material to fall freely, off the conveyor end. Where discharge chutes deliver to other apparatus or devices and are liable to become choked, the material will pack up and eventually reach the discharge head pulley. In such instances sufficient clearance should be allowed between the chute top plate and the belt passing round the pulley to allow the largest pieces to pass over, thus preventing damage to the conveyor belt. It is a good insurance to equip the discharge chute or bin with a limit or pressure switch to stop the movement of the conveyor belt. This would save both damage to the belt and overflowing of the material.

Renewable liner plates are very often used. For coke, ash and similar abrasive materials cast-iron plates are used. For sand, gravel, ore, etc., $\frac{3}{8}''$ M.S. plates are used. Provision should be made for clearance and storage of tramp iron when a magnetic separation pulley is used.

Trajectory of Material Over Discharge Pulley.

To reduce wear on the chute and breakage of fragile material passing through it the location and design should also take into consideration the path or trajectory that the material will tend to follow when discharged over a pulley. This path, which is determined by the combined influence of gravity, belt speed and pulley diameter, can be found graphically from Fig. 53.



D	E	F	G	H	J	K	L	M	N	P	Q	R	S	T
$\frac{1}{2}$"	$1\frac{15}{16}$"	$4\frac{11}{32}$"	$7\frac{23}{32}$"	$12\frac{1}{16}$"	$17\frac{3}{8}$"	$23\frac{5}{8}$"	$30\frac{7}{8}$"	$39\frac{1}{16}$"	$48\frac{1}{4}$"	$58\frac{3}{8}$"	$69\frac{15}{16}$"	$81\frac{17}{32}$"	$94\frac{9}{16}$"	$108\frac{1}{2}$"

Fig. 53.

A is the angle from vertical at which the material will leave the belt as it travels over discharge pulley. It is found at the point on the chart where a line projected to the right from the belt speed intersects a line projected downward from the pulley diameter. If the line of intersection is above the zero line, method No. 1 should be used to determine the trajectory except when angle A is less than angle W. Then method No. 2 should be used.

If the point of intersection is below zero degree line, then A = 0 and the trajectory can be determined by methods 2, 3 and 4 where the tangent line is a continuation of the direction of travel of the belt approaching the pulley.

B = 1 inch per 100 feet per minute speed of belt (4″ for 400 F.P.M., $2\frac{1}{2}$″ for 250 F.P.M., etc.) measured along tangent line at same scale used for indicating dimension C.

C = radius of discharge pulley plus 1 inch (19 inches for 36 inches diameter pulley, 25 inches for 48 inch pulley, etc.). This 1 inch added to the pulley radius is intended to represent the approximate thickness of the belt and therefore the lower particles of the material. The chart is based on this value but if it should be desired to find A where C is different, then

$$\text{Cos } A = \frac{V^2}{GC}$$

Where V = belt speed in feet per second.

G = acceleration of gravity 32·2.

C = distance in feet from centre of pulley to trajectory line desired.

ANTI-RUN BACK DEVICES.

When the unbalanced gravitational forces in a conveyor belt exceed the frictional force, as in the steeper ascending conveyors and in conveyors having steep ascending portions, a hold back or anti-run back gear should be provided to prevent loaded conveyors from running backward when they are stopped or power is interrupted. If a loaded or practically loaded conveyor should run backward considerable material may accumulate around the lower end to be cleared away and reloaded, possibly causing considerable damage to the belt and other parts of the conveyor. Anti-run backs are usually selected on the basis of pounds feet of torque to be resisted.

Knowing the gravitational and frictional forces in the conveyor as found from formula (Table No. 44), the foot pounds of torque required to be held by a run back gear placed on the drive pulley shaft can be calculated from the following formula :—

$$T = \left(G - \frac{F}{2}\right) \quad \left(\frac{R}{12}\right)$$

Where F = total frictional forces opposing movement of belt.

G = nett or unbalanced gravitational forces causing backward movement of belt.

R = radius of conveyor drive pulley in inches.

T = foot lbs. torque required at hold back placed on drive pulley shaft.

If the H.P. required to operate the conveyor has been determined the approximate pounds feet torque required to be held by the hold back placed on the drive pulley shaft can be calculated from the following formula :—

$$T = \left(\frac{275OR}{S}\right) \left(H_1 - \frac{H_2 + H_3}{2}\right)$$

Where H_1 = H.P. to elevate material.

H_2 = H.P. to transport material horizontally.

H_3 = H.P. to operate empty conveyor.

R = radius of conveyor drive pulley in inches.

S = speed of belt in feet per minute.

T = pounds feet torque required of hold back placed on drive pulley shaft.

The foregoing formulas do not take into consideration the friction in the pulley shaft bearings or the drive equipment which will assist in preventing a conveyor from running backwards. It is recommended that they be disregarded and the hold back be placed on the drive pulley shaft so that replacement of any of the drive equipment can be made without having to unload the conveyor.

If the hold back were placed on a faster turning countershaft, its required holding torque would be inversely proportioned to the speed of the pulley shaft and the shaft on which it is placed, thereby permitting a smaller hold back, but this is not recommended except where it may be justified because of space limitations or supporting difficulties.

Calculations should be made considering the various ways the conveyor may be loaded when power may fail or be shut off, so that the hold back will be large enough to take care of the worst conditions. For instance, with a conveyor having a horizontal run leading into a steep ascending run, the worst conditions would be when the conveyor is stopped with no material on the horizontal belt and the inclined belt is fully loaded.

A simple, effective and dependable hold back is the band brake type. It is rugged, dependable and quiet in operation, a slight reverse motion of the shaft causing it to function with a clamping effect. When used with an enclosed gear drive, such as a worm reduction gear, the hold back can be fitted on the motor shaft between the flexible coupling and the motor. This position permits a smaller size brake.

The following list gives the maximum operating speed for each size of brake wheel.

TABLE No. 54.

Brake wheel diameter in inches.	Torque in lbs. feet.	Maximum r.p.m.
$8\frac{1}{4}$	300	1800
12	1200	1800
18	3150	1200
24	8500	900
30	16500	720

The foregoing type of hold-backs are listed on the basis of torque in pound feet. The torque required at the head shaft to prevent the conveyor running backwards equals the operating torque, less friction. Knowing this, the following formula may be used to determine the torque required.

$$T = \frac{t \times s}{R}$$

If the H.P. of the prime mover is known, the torque can be calculated as follows :—

$$T = \frac{5250\,H}{R}$$

Where T = torque on brake wheel in lbs. ft.
t = torque of head shaft in lbs. ft.
S = speed of head shaft in revolutions per minute.
R = speed of brake wheel in revolutions per minute.
H = H.P.

A protective housing should be provided when the hold back is subjected to extreme dust or climate conditions. Another type of hold back is the positive ratchet hook. On the friction type, the pawl is drawn into mesh with the rachet wheel by means of spring tension clamps acting on its hub when the rachet wheel reverses. The pawl is clear of the rachet wheel except when the direction of rotation is reversed.

On the solenoid type the pawl is held disengaged, clear of the ratchet teeth, by a solenoid. The instant the current is off, the pawl is forced into engagement with the rachet teeth.

On either of these two types the action is so quick that the conveyor is positively stopped before it has gained momentum, thus preventing damage or undue shock to the machinery.

The solenoid type is specially recommended for installation where a dusty condition prevails or where there are frequent current interruptions.

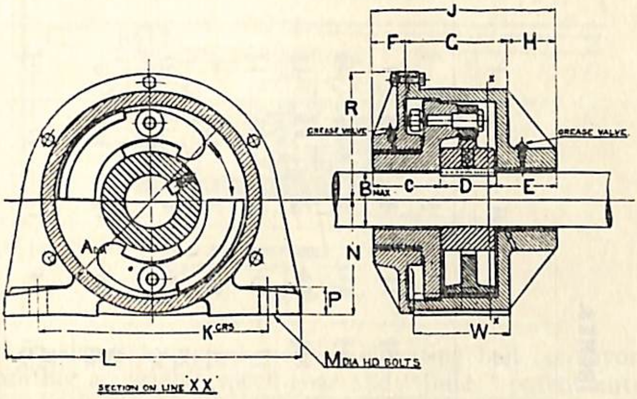


Fig. 54.

The Jones automatic brake or backstop, as shown in Fig. 54, is manufactured by Crofts, Ltd., Bradford.

This brake consists of five main parts :—

(1) The shell.
(2) The wedge block.
(3) Flanged hub on which are pivoted the brake shoes.
(4) The brake shoes.
(5) Dust proof end cover.

When installed the complete brake is fixed on the conveyor drive shaft and securely fixed to the structural supporting steelwork, to prevent rotation, the wedge block being keyed to the conveyor shaft.

In the normal direction of running the one way member engages with lugs on the brake shoe, thus carrying round the flanged hub with the shaft. When the conveyor stops from any cause and reverses direction, the one-way member slides in the curved path of the brake shoe, forcing the latter outwards with increasing pressure on the inner surface of the shell, thus arresting reverse rotation without shock. Engagement and release is entirely automatic at all times.

The power of the brake is listed as equal to the equivalent shaft diameter E.

TABLE No. 55.
APPROXIMATE DIMENSIONS IN INCHES.

Size of Coupling.	A	Max. B	C	D	E	F	G	H	J	K	L	M	N	P	R	W
1	6½	1	2⅛	1⅜	1⅝	1⅛	2¾	1¼	5⅛	7	9	½	3½	¾	4	2⅝
2	8	1½	2⅝	1½	2	1½	3⅛	1½	6⅛	8¾	11	⅝	4¼	1	4⅞	3
3	10½	2	3½	2	2½	2⅛	4	1⅞	8	11½	14½	⅝	5½	1¼	6⅜	3¾
4	13½	3	5⅝	3	4	3¼	6⅛	3¼	12⅝	14½	18	¾	7¼	1⅜	8¼	5¾
5	16	4	6⅛	4	5	3⅝	7¼	4¼	15⅛	17	21	⅞	8½	1½	9½	6¾
6	19½	5	6¾	5	5½	4¼	8⅝	4⅜	17¼	21	26	1	10½	2	11⅜	8
7	24	6	7½	6	6	4½	10⅜	4⅝	19½	26 x5	32	1	13	2½	13¾	10
8	27	8	9	7½	6½	5	13	5	23	29 x6	36	1¼	14¼	3	15¾	12
9	32	10	11	8	8	6⅜	14⅝	6	27	35 x7	44	1½	17	3½	18½	13½

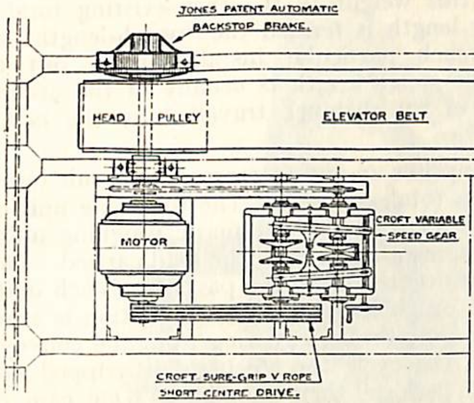


Fig. 55.

Fig. 55 shows a typical inclined elevating belt conveyor drive incorporating a variable speed gear and "Jones" patent automatic backstop brake or run back. It will be seen that the brake is securely fixed to the conveyor supporting framework. The brake is of strong but simple construction and there is practically no possibility of mechanical trouble. It requires no adjustment and only periodical lubrication and cleaning.

WEIGHING OF MATERIAL IN TRANSIT.

It is often necessary to weigh and record the exact amounts of material handled by various conveyors, as a check on volumes delivered to different departments and the total tonnage loaded in or out of a plant. With the automatic weigher and totaliser the material can be weighed as it passes along on the conveyor in ordinary transit to $+ \frac{1}{2}\%$ of accuracy.

THE "BLAKE-DENISON" WEIGHER AND TOTALISER.

The "Blake-Denison" weigher and totaliser has been used on conveyor systems for over 50 years, and being approved by weights and measures authorities it is usable for trade purposes in most parts of the world.

The weigher will operate on horizontal conveyors or inclined conveyors where the angle of inclination is not so great as to cause roll back of the material.

The principle of operation is that of dividing the travelling belt into units of length, weighing each unit in turn as it comes along, and adding this weight to the then-existing total on the totaliser. The unit length is termed the "weigh-length" and has to be settled for each particular installation to suit the site characteristics. The weigh-length is usually of the order of one sixth to one-tenth of the distance travelled by the belt in one minute.

The machine comprises an automatic weighing unit terminating in a steelyard, and a totalising unit. The totalising unit operates as follows. The steelyard of the automatic weighing unit moves from its zero position in proportion to the load carried. One cycle of the totalising unit occurs during the passage of each unit length of the belt over the weigh length. Synchronisation is ensured by driving the totaliser unit through gearing from the conveyor belt. At a fixed point in the cycle the steelyard is gripped and held, whilst certain cams perform their functions. One cam operates a quadrant, moving it forward until it makes contact with the steelyard ; the angular movement of the quadrant is thus a measure of the displacement of the steelyard from its zero position. On its return to its home position the quadrant drives, through suitable mechanism, the totaliser. This drive, incorporating discount means (described later) is a positive one. It consists of three members free to rotate on a horizontal spindle ; the outer ones are fixed laterally and the centre one is movable laterally by an arm so that it may be connected to either one of the outer members. The centre member has limited-motion spring-urged pawls on its inner face. The inner member is moved axially only when it and both outer members have no angular velocity. By this means wear of pawl and ratchet teeth is rendered negligible. The quadrant is attached to one outer member ; the other outer member is a discount motion allowing the steelyard to move below zero to cater for variations in tare. Hence, whilst the steelyard is moving to a position counter-balancing the load the discount motion is clutched in. The quadrant member is turned to measure the displacement of the steelyard and after that the quadrant member is clutched to the inner member (thus de-clutching the discount mechanism) and the quadrant member is then returned to "home" and in doing so turns through suitable gearing the totaliser. The steelyard is then released to take up its position as dictated by the load on the next unit of belt.

The mechanism can be arranged to operate two totalisers if necessary, and a ticket stamping totaliser can be supplied. Distant indicators, totaliser or recorders can be used where desired ; and a rate of flow indicator, operated from the steelyard, is available. Should the conveyor be reversible the weigher can be arranged to weigh for either one or both directions of travel.

In positioning the weigher on a conveyor the following points should be observed. As the tension in the carrying side of the belt

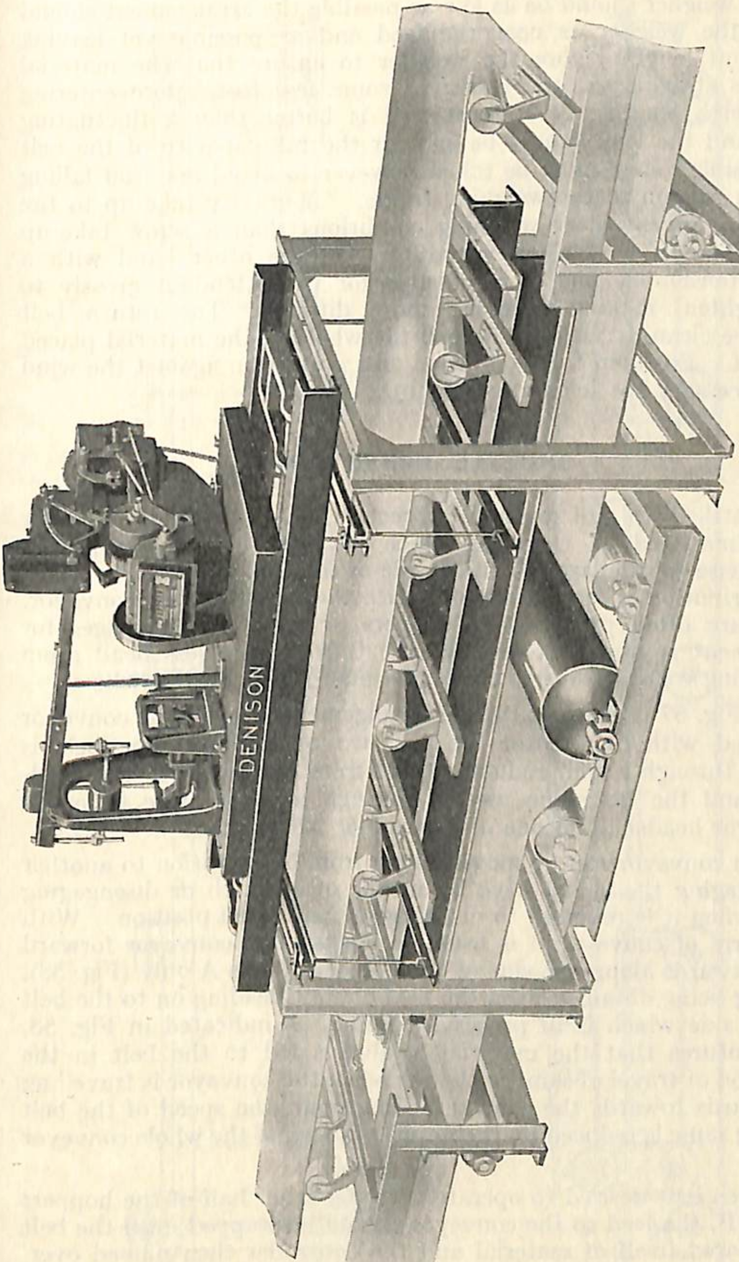


Fig. 56.

at the weigher should be as low as possible the arrangement should place the weigher as near the feed end as possible yet leaving sufficient length before the weigher to ensure that the material reaches stability on the conveyor some few feet before entering the weigh length. A uniform feed is better than a fluctuating feed, and the load should be as near the full capacity of the belt as possible ; care must be taken however to avoid material falling off the belt on to the weighing frame. A gravity take-up to the conveyor gives better weighing conditions than a screw take-up since there is no tension variation. On the other hand with a screw tensioning gear it is possible for the attendant grossly to overtighten, making weighing more difficult. The return belt must be clean, having discharged the whole of the material placed upon it. Freedom from vibration and protection against the wind are necessary for accurate weighing.

SHUTTLE CONVEYORS.

Shuttle belts are reversible direction, self-contained, conveyor units, mounted on carriages which permits them to be moved lengthwise to discharge to either side of the feed point. The range of distribution is approximately twice the length of the conveyor. They are often preferred to trippers or throw off carriages for permanent storage systems, because they require less head room and being without reverse bends are much easier on the belts.

In Fig. 57 is shewn a typical arrangement of a shuttle conveyor arranged with one motor for the two operations. The belt is driven through a spur reduction gear from the motor to the headshaft and the travelling motion through a chain drive from the conveyor headshaft to one of the runner wheel axles.

The conveyor can be moved along from one position to another by engaging the chain drive by means of a clutch or disengaging same when it is required to discharge at any fixed position. With this form of conveyor it is usual to operate the conveyor forward or backwards along one side of the line of hoppers A only (Fig. 58), loading being obtained from the feed chute C leading on to the belt at the side which is in process of filling, as indicated in Fig. 58. This ensures that the material is always fed to the belt in the direction of travel of same, although when the conveyor is travelling backwards towards the central feeding point the speed of the belt passing same is reduced by the returned speed of the whole conveyor frame.

When it is desired to operate over the other half of the hoppers or pile B, the feed to the conveyor should be stopped until the belt has cleared itself of material and the controller then moved over, thus reversing the direction of travel of the belt. Feeding can

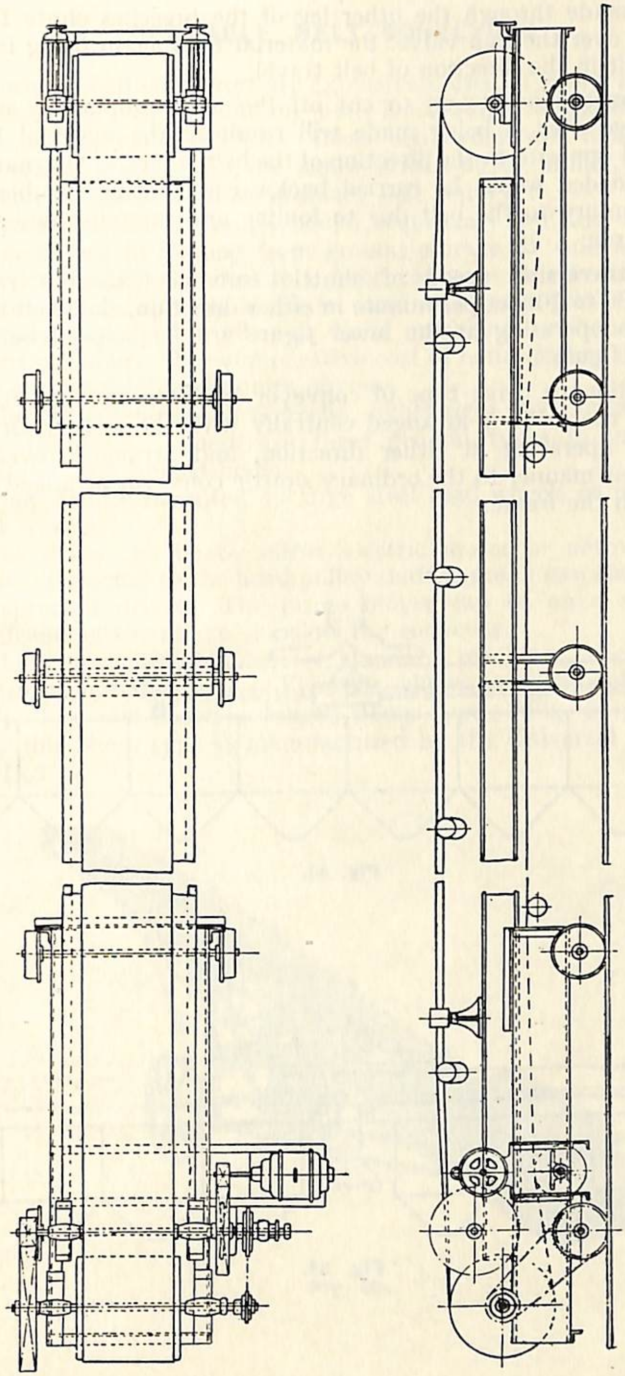


Fig. 57.

then be made through the other leg of the breeches chute D, by swinging over the flap valve, the material thus again being fed on to the belt in the direction of belt travel.

Precautions not taken to cut off the feed temporarily whilst the change over is being made will result in the material being either fed opposite to the direction of the belt travel, or the material already loaded would be carried backwards, causing trouble and perhaps injury to the belt due to fouling and jamming under the skirt boards.

The traversing speed of shuttle conveyors usually ranges between 30 to 70 feet per minute in either direction, slow belt speed conveyors operating at the lower figure and high-speed belts at the high figure.

A stationary drive type of conveyor is shewn in Fig. 59. In this case the drive is arranged centrally with the feed shoot, the conveyor operating in either direction, and arranged reversible in a similar manner to the ordinary shuttle conveyor arranged with a drive on the frame.

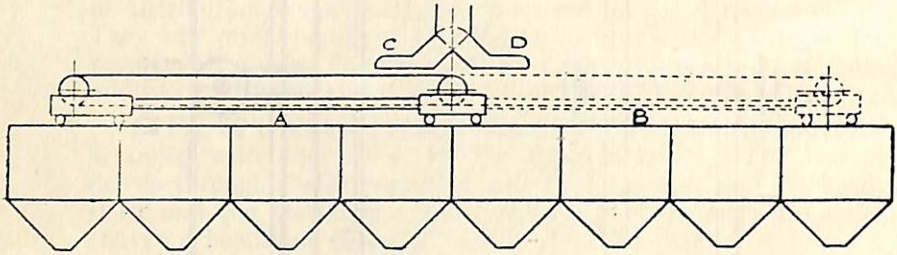


Fig. 58.

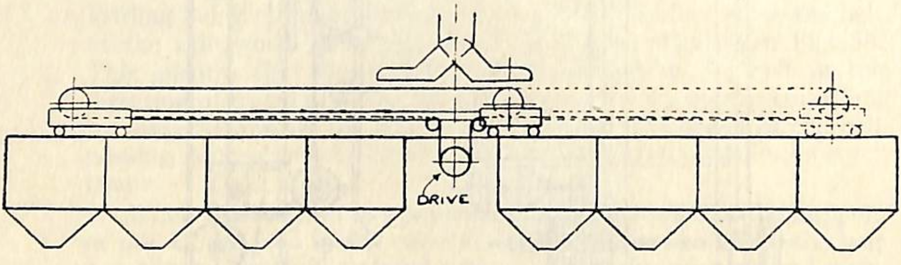


Fig. 59.

PORTABLE BELT CONVEYORS.

Portable belt conveyors are extensively used in many industries to reduce the cost of handling both loose bulk and packages over short distances or heights. These conveyors will handle coal, sand, gravel, stone, grain, ore, ashes, bricks, boxes, in fact, anything that can be handled on an ordinary belt conveyor.

In adaptations to the particular requirements of the user, they find great use in loading from ground storage to railway trucks or lorries, feeding material to a fixed conveyor or elevator from a variable source, such as a pile of sand or storage pile. For such materials as coal or sand, etc., these machines can handle up to 60 tons per hour. The comparative cost of hand loading as against using such a machine is quite obvious.

The design should be powerful, lightweight, self-contained and easily portable. There are three general types in use, fixed, adjustable and hinged boom.

They can be mounted on large steel road wheels or pneumatic tyred wheels.

The drive can be by either electric motor or petrol-paraffin engine, connected to the head pulley shaft through two chain drives and sprocket drives. The prime mover can be on a structural steel frame either above or below the conveyor.

There are many makers of standard machines, each having their own specifications. Fig. 60A shows an adjustable boom type; Fig. 60B shows a hinged boom type; Fig. 61 shows an adjustable boom type as manufactured by the Universal Conveyor Co., Ltd.

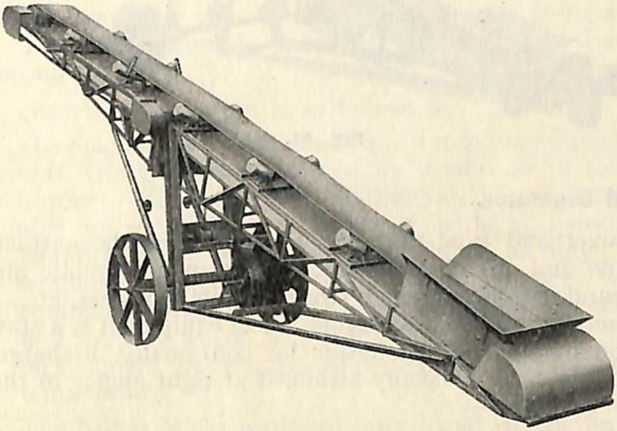


Fig. 60a.

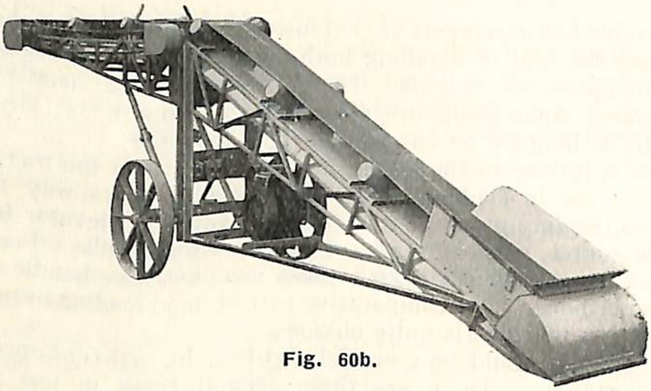


Fig. 60b.

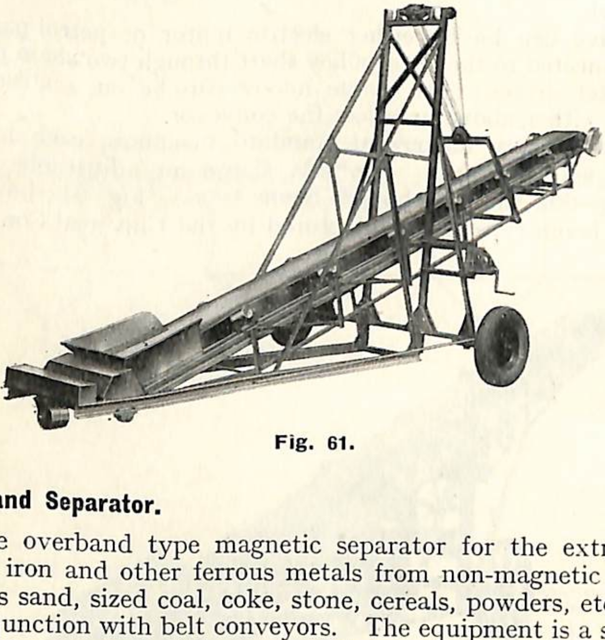


Fig. 61.

Overband Separator.

The overband type magnetic separator for the extraction of tramp iron and other ferrous metals from non-magnetic materials such as sand, sized coal, coke, stone, cereals, powders, etc., is used in conjunction with belt conveyors. The equipment is a stationary suspension magnet with a means for continually discharging the extracted iron. It is usually arranged at right angles to the main conveyor because :—

(*a*) A magnetic barrage from a North and South pole is arranged at right angles to the flow of material, through which the iron must pass. This is the most effective means of magnetising and extracting ferrous materials.

(*b*) By installing an overband separator across the main conveyor it is usually convenient to keep the magnet clear of the majority of structures.

(*c*) A small unit can be designed to accomplish the separation.

If the separator is arranged in line with the main conveyor, the makers point out the following disadvantages :—

(1) The magnetic unit must be longer to carry the extracted iron well over the delivery end of the head pulley.

(2) The head pulley should be of non-magnetic material, otherwise this becomes magnetised by induction and will tend to hold small pieces of iron down rather than allow them to move upwards to the magnet face.

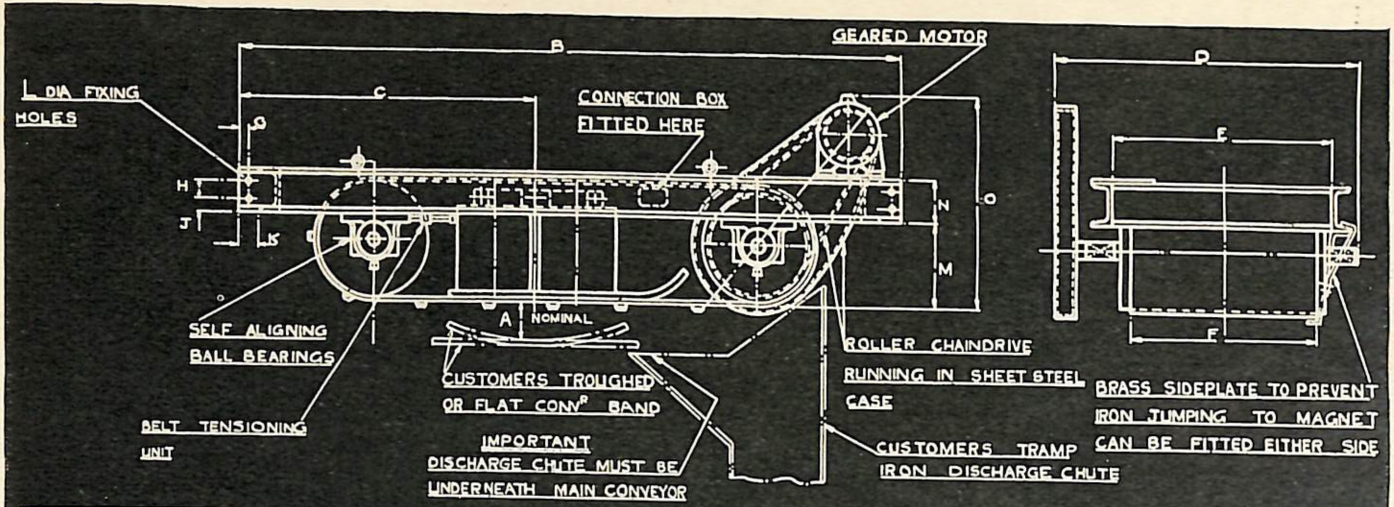
(3) When extracting small iron in the form of grits, brads, and pellets from foundry sand, the overband separator belt can be arranged at any convenient speed. When extracting long pieces of iron, however, such as core wires, it is necessary that the overband belt move faster than the main belt. This is because the front end of the rod or bar will be lifted to the magnet and if the main belt is faster than the separator, it will tend to push the back end of the bar forward causing the bar to dig into the main conveyor and jam. The distance through which the iron is to be lifted is important. The manufacturers must know this as the size of the magnet unit will be dependent upon the depth of field necessary to attract and lift the iron from the main belt.

Fig. 62 shows the general overall dimensions and particulars of the "Boxmag" magnetic separator, made by Electromagnets, Ltd., Birmingham.

The general construction is as follows :—

(*a*) The magnetic unit consists of a soft iron forged steel magnet unit whilst the coils are usually wound on to solid metal formers. They are continuously rated, and treated suitable for normal conditions met with in foundries and factories. Any special condition should be noted, and the windings treated accordingly. For instance, when the equipment is installed almost on top of the knock-out grid in a foundry where heat and steam is met with, the windings should be treated to withstand the humidity and temperature.

(*b*) The belt is of the ordinary canvas and rubber type, unless specially desired otherwise, and this belt is fitted with low


L DIA FIXING HOLES
B
C
G
H
J
K
GEARED MOTOR
CONNECTION BOX FITTED HERE
D
E
F
N
O
M
A
NOMINAL
SELF ALIGNING BALL BEARINGS
BELT TENSIONING UNIT
CUSTOMERS TROUGHED OR FLAT CONVR BAND
IMPORTANT DISCHARGE CHUTE MUST BE UNDERNEATH MAIN CONVEYOR
ROLLER CHAINDRIVE RUNNING IN SHEET STEEL CASE
CUSTOMERS TRAMP IRON DISCHARGE CHUTE
BRASS SIDEPLATE TO PREVENT IRON JUMPING TO MAGNET CAN BE FITTED EITHER SIDE

REF SIZE No	WIDTH OF MAIN CONVEYOR	DIMENSIONS														CONSUMPTION IN WATTS.	H.P. OF MOTOR	NETT. WT. IN LBS.	GROSS WT. IN LBS.	SHIPPING SPACE IN CUBIC FT.
			B	C	D	E	F	G	H	J	K	L	M	N	O					
O.B. 2414	12" TO 14"	DIMENSION A. 2½"	5'-11"	2'-7"	3'-2"	2'-4"	2'-0"	1¼"	2"	1½"	3"	9/16"	9⅝"	5"	1'-11½"	460	½	1100	1350	79
O.B. 2418	16" TO 18"		6'-1"	2'-8"	3'-2"	2'-4"	2'-0"	1¼"	2"	1½"	3"	9/16"	9⅝"	5"	1'-11½"	580	½	1300	1580	82
O.B. 2424	20" TO 24"		7'-0"	3'-1½"	3'-2"	2'-4"	2'-0"	1¼"	2"	1½"	3"	9/16"	9⅝"	5"	1'-11½"	990	½	1600	1950	91
O.B. 2430	28" TO 30"		7'-3½"	3'-3½"	3'-2"	2'-4"	2'-0"	1¼"	2"	1½"	3"	9/16"	9⅝"	5"	2'-1½"	1300	¾	2000	2450	94
O.B. 3018	16" TO 20"	DIM. A. 4"	6'-11"	2'-11"	3'-10"	2'-11"	2'-6"	1¼"	3"	1½"	3"	9/16"	11"	6"	2'-3"	880	½	1800	2350	105
O.B. 3024	20" TO 24"		7'-5"	3'-2"	3'-10"	2'-11"	2'-6"	1¼"	3"	1½"	4	13/16"	11"	6"	2'-3"	1000	¾	2200	2850	130
O.B. 3036	30" TO 42"		9'-1"	4'-0½"	4'-0"	2'-11"	2'-6"	2"	3"	1½"	4"	13/16"	11"	6"	2'-5"	1580	1	2600	3350	145.
O.B. 3624	20" TO 24"	DIM. A. 6"	8'-2"	3'-5"	4'-5"	3'-5"	3'-0"	2"	3"	1½"	4"	13/16"	13"	6"	2'-7"	1200	1	2350	3050	150
O.B. 3630	24" TO 32"		8'-8"	3'-8"	4'-5"	3'-5"	3'-0"	2"	3"	1½"	4"	13/16"	13"	6"	2'-7"	1550	1	3000	3750	170
O.B. 3636	34" TO 42"		10'-2"	4'-5"	4'-5"	3'-5"	3'-0"	2"	3"	1½"	4"	13/16"	13"	6"	2'-8"	2175	1¼	4150	5050	195
O.B. 4230	24" TO 32"	DIM. A. 8"	9'-10½"	4'-2"	5'-0"	4'-0"	3'-6"	2"	3"	1½"	4"	13/16"	15½"	6"	2'-10"	2020	1¼	4300	5100	220
O.B. 4236	34" TO 42"		10'-4½"	4'-6½"	5'-0"	4'-0"	3'-6"	2"	3"	1½"	4"	13/16"	15½"	6"	2'-11"	2900	1½	5400	6500	230

Fig. 62.

retentivity Lowmoor cast-iron magnetic feelers which intensify the magnetism and the influence of the magnetic unit, and also form mechanical aids to push the extracted material away from the magnetic unit.

(*c*) The magnet units have carrying poles which are turned up at the end. This is designed so as to gradually reduce the magnetic pull and allow the iron to fall by gravity. If this were not arranged in this way, the magnetic strength would be such that the iron would continually jump back to the position underneath the poles. (See also magnetic separator pulleys).

DRIVES FOR BELT CONVEYORS.

The drive should be designed for peak loads at the fastest speed and narrowest belt width which will satisfactorily handle the material and capacity required without breakage or poor loading conditions. Lighter and more economical belts can be used at reduced tensions.

Every drive should be located as near the head end as possible, although comparatively short conveyors may be driven near the tail end without undue belt tension.

Select head and tail pulleys from tables given, keeping the diameters as large as possible to minimise bending belt and stretching of outer plies. Shafts of proper diameter can also be calculated from the formulas given.

In short and medium length conveyors, the drive pulleys get sufficient traction without the use of snub pulleys. Longer conveyors require greater "effective" belt tension, which is obtained by :—

(1) Increasing initial or slack side tension.

(2) Lagging the plain drive pulley with a covering of rubber.

(3) Using a snub pulley to increase the arc of contact.

(4) Using a tandem or two pulley drives to further increase the arc of contact.

(5) Using a pneumatic pressure pulley drive.

Single Pulley Drives.

As shown in Fig. 39, the simplest form of drive consists of a bare pulley and belt with an arc of contact of approximately 180°. As the conveyor length is increased, the driving grip of the pulley must be increased by maintaining a greater tension in the belt. If the driving tension added to this initial tension brings the total belt tension close to the ultimate allowable tension for the belt chosen, another type of drive must be used, or a belt with more plies used to permit greater tension. Using a rubber lagged pulley will increase the driving capacity of the same belt from 3 to 10% and the lagging reduces the abrasion between belt and pulley. The lagging is usually vulcanised on the pulley.

The application of a snub pulley just back of the head pulley will increase the arc of contact from approximately 180° to 230° and increase the tractive effort with a reduced slack side tension in the belt. A snub pulley also relieves the strain on the first return idler back of the drive, but should be used as sparingly as possible because of the added flexing from the reverse bending in the belt. Whenever a snub or second drive pulley becomes necessary, care must be taken not to let damp or sticky material build up on the snub pulley. This can be prevented by the use of a spring type belt cleaner, a power driven belt brush or the installation of a slatted type pulley.

Tandem Drives.

Longer conveyors and those elevating the material to a considerable height often cannot be driven by a single pulley. Tandem drives permit the necessary increase in arc of contact to a total of from 420° to 480° and a well-designed drive will give 25% greater "effective tension" without increasing the total tension in the belt. In this type of drive, the main pulley carries approximately 75% and the auxiliary pulley 25% of the load. Both pulleys should be accurately machined and lagged to a uniform thickness to prevent unnecessary slipping on one of the pulleys. (See Fig. 39).

The heavier rubber cover on the carrying surface of the belt may cause slight slippage, due to the unequal effective belt diameter. This is compensated by making one pulley slightly larger or by means of a differential drive. As the rubber surface wears, this difference gradually decreases and any difference diameters should be based on an average thickness.

Records of long belt life from conveyors with geared tandem drives, indicate that the more complicated differential drive is seldom of great advantage except for conveyors of unusual size and length.

DRIVE ARRANGEMENTS.

To transmit rotational energy from a source of power to a point of application, many forms of drives can be used, either a single drive or a combination of drives, depending upon the reduction in speed required.

Silent Chain Drives are used for smooth, dependable and positive transmission of power including heavy loads at high speeds.

Roller Chain Drives are used for heavy loads at moderate speeds and moderate loads at high speeds.

V Belt Drives provide efficient transmission of power at high speeds when temperatures are moderate.

Variable Speed Drives provide a positive speed change and precise adjustment to an infinite number of speed variations, control is easy and changes can be made while drive is in operation. The P.I.V. variable speed drives embody a unique tooth forming chain, operating between grooved conical wheels in an enclosed housing.

Fluid Drives employ oil as a medium for transmitting power from the drawing to the driven shafts. They provide smooth easy starting and automatic overload protection as described under fluid couplings.

Geared Motors comprise helical gear sets in an enclosed housing with integrally mounted motors for simplicity and compactness.

Herringbone and Helical Gear Drives comprise single, double and triple sets of gears in rugged enclosed housings. They provide efficient, smooth running transmission of power in heavy duty industrial service.

Worm Gear Drives employ worm and gear sets singly, doubled or in combination with helical gear sets to provide high ratio transmission of power. The single worm and helical worm gear drives are used for right angle applications. They are sturdy and compact and give quiet and dependable service.

A simple form of drive is the all-gear drive consisting of countershaft with two sets of spur gears, or one set of gears for the final drive and a chain or V belt from the motor to the countershaft.

Many of the foregoing drives singly or in combination can be used depending upon the particular requirements of the conveyor under review. See Fig. 55.

Fluid Couplings.—The main advantages of incorporating a "*Fluidrive*" in the drive as follows :—

"*Fluidrive*" permits the motor to start up quickly against light load, even if the conveyor is overloaded.

Permits the motor to develop its peak torque for accelerating the load.

Take up of the drive is the smoothest possible and the rate of acceleration can be adjusted in the traction coupling by varying the oil filling.

The torque transmitted can be limited to a pre-determined value even if the drive is suddenly stalled.

The motor can continue to run for a period whilst the conveyor is stalled without loss of torque.

"Fluidrive" enables more favourably starting and running conditions for the motor resulting in a simplification of the electrical equipment.

Squirrel cage motors and direct on starters are fundamentally the simplest and most reliable type of electrical equipment and *"Fluidrive"* makes a definite contribution to their use by virtue of its special characteristics.

Large numbers of belt conveyors drives are now fitted with *"Fluidrive"* for the main reason that a standard squirrel cage motor can be rated for the running instead of the starting condition, particularly when, due to low temperature or other reasons, the starting torque is extra heavy.

The Fluidrive Coupling and Engineering Co., Ltd., Isleworth, makers of the *Vulcan-Sinclair Fluid Coupling Unit,* give various examples of the installation of both traction and scoop control fluid couplings.

STARTING AND STOPPING.

When starting up, the conveyor should have attained its full speed before any material is fed on to the belt, and when stopping the conveyor should be run at full speed for a few minutes after the feed has been stopped, in order to remove all material.

With a series of conveyors it is advisable to run the whole system for five or ten minutes after the feed has been stopped in order to have delivered the laggards or those pieces which may have a tendency to roll back and which if delivered over the head pulleys when going slow, join in the transfer chutes and thereby cause great injury to the belt.

Frequent starting and stopping under full load is apt to cause belt stretch and the plies to be pulled out, or the belt may break at the joint, causing endless trouble and expense.

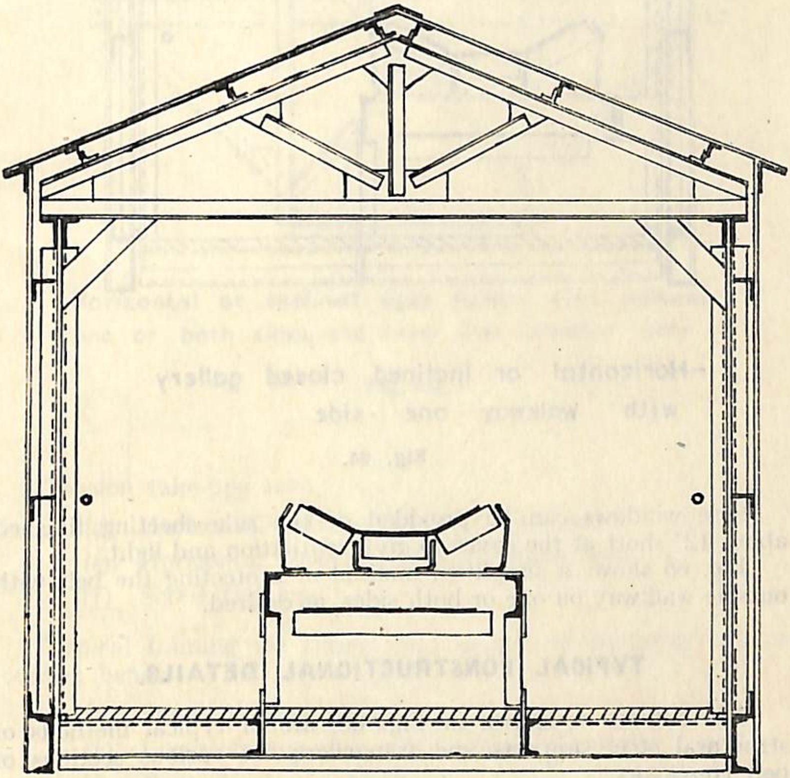
A system now being adopted on some plants where several elevators or conveyors are arranged is the inter-locking of each conveyor motor control panel, arranged with push-button stops located at intervals along each conveyor run. By this means all the conveyors preceding the one which it is required to stop are also electrically stopped, but the following units will continue to operate thus discharging their contents.

Arrangements can also be made on each of the control panels by means of isolating switches, so that any individual or any set of conveyors can be operated for overhauling or inspection.

TYPICAL GALLERIES AND HOUSING.

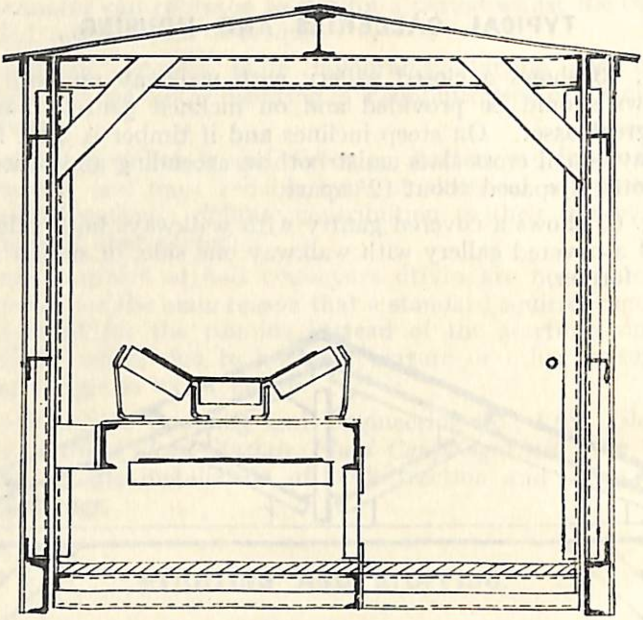
Fig. 63 shows a closed gallery with walkway on both sides. Windows should be provided and on inclined gantrys handrails are a great asset. On steep inclines and if timber is used for the walkway, small cross slats assist both in ascending and descending the gantry if spaced about 12″ apart.

Fig. 63 shows a covered gantry with walkways both sides, and Fig. 64 a covered gallery with walkway one side, of similar design to the former.



-Horizontal or inclined closed gallery with walkway both sides

Fig. 63.



-Horizontal or inclined closed gallery with walkway one side

Fig. 64.

Side windows can be provided or the side sheeting finished about 12″ short at the eaves to give ventilation and light.

Fig. 65 shows a simplified method of protecting the belt with outside walkway on one or both sides, as desired.

TYPICAL CONSTRUCTIONAL DETAILS.

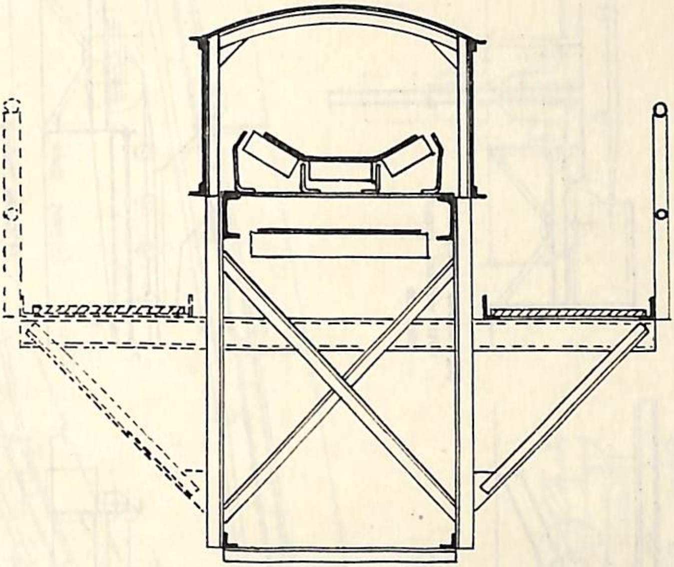
The details, as shown in Fig. 66, are of typical methods of structural steel supports and framework for various sections of belt conveyors.

Head terminals are :—

(*a*) Snubbed pulley drive.

(*b*) Tandem drive.

(*c*) Snubbed pulley drive with vertical gravity tension take-up.



- Horizontal or inclined open gallery with walkway one or both sides, and cover over conveyor only

Fig. 65.

Tension take-ups are :—

(*d*) Horizontal gravity take-up above floor.
(*e*) Horizontal gravity take-up through floor.
(*f*) Screw take-up.

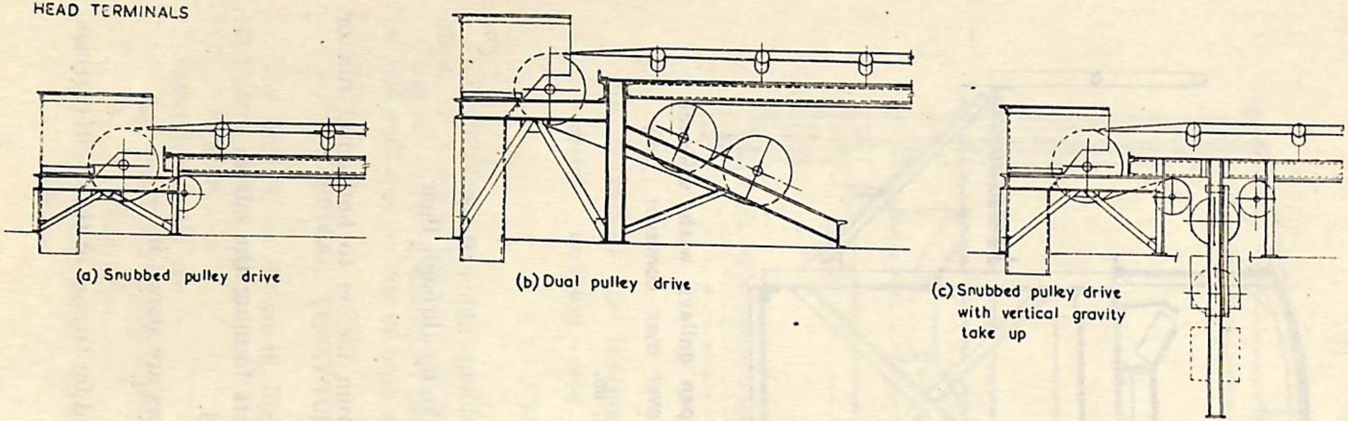
Typical framing for bends from incline to horizontal run or convex bends :—

(*g*) Shows a bend pulley.
(*h*) Incorporating idler sets forming the curve.
(*j*) Typical concave bend.

Cross sections of belt conveyors are shown in :—

(*k*) Plain belt conveyor.
(*l*) Belt conveyor arranged for tripper or throw-off carriage.

HEAD TERMINALS

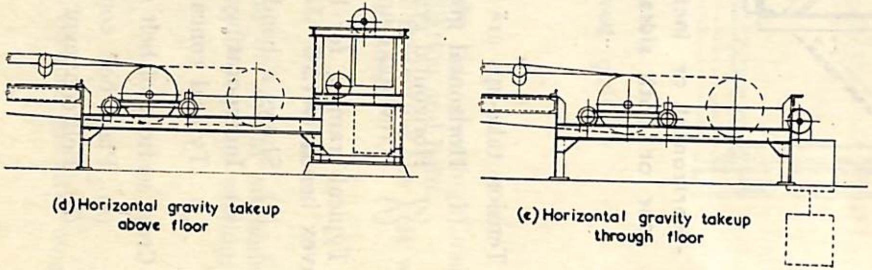


(a) Snubbed pulley drive

(b) Dual pulley drive

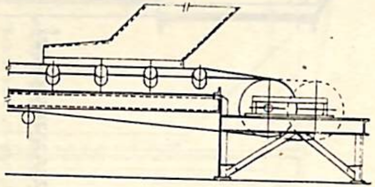
(c) Snubbed pulley drive with vertical gravity take up

Fig. 66.



(d) Horizontal gravity takeup above floor

(e) Horizontal gravity takeup through floor



(f) Screw takeup

Fig. 66a.

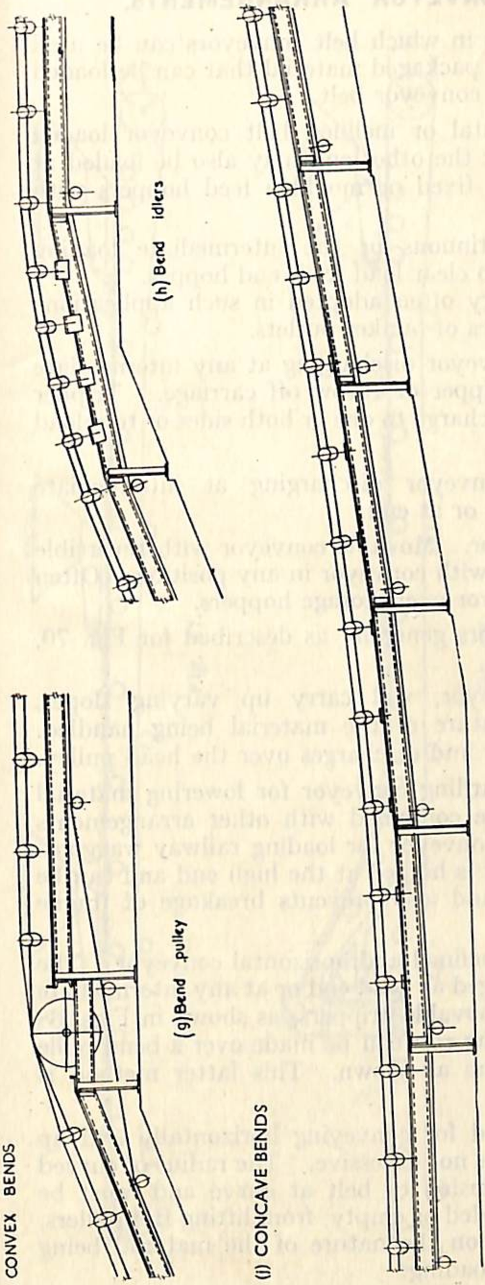


Fig. 66b.

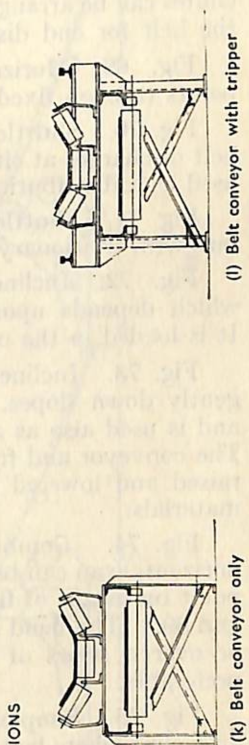


Fig. 66c.

TYPICAL BELT CONVEYOR ARRANGEMENTS.

A few of the many ways in which belt conveyors can be used to carry any kind of bulk or packaged material that can be loaded onto or discharged from the conveyor belt.

Fig. 67 shows a horizontal or inclined belt conveyor loaded at one end, and discharge at the other end may also be loaded at intermediate points through fixed or movable feed hoppers with chutes.

Skirtboards can be continuous or the intermediate loading hopper skirt boards hinged to clear load from end hopper.

Vibratory feeders are very often adopted in such applications where the feed is from a series of bunker outlets.

Fig. 68. Horizontal conveyor discharging at any intermediate point through a movable tripper or throw off carriage. Tripper chutes can be arranged to discharge to one or both sides or to reload the belt for end discharge.

Fig. 69. Horizontal conveyor discharging at intermediate points through fixed trippers or at end.

Fig. 70. Shuttle conveyor. Movable conveyor with reversible belt discharges at either end with conveyor in any position. Often used as a distributing conveyor over storage hoppers.

Fig. 71. Shuttle conveyors generally as described for Fig. 70, but with stationary drive.

Fig. 72. Inclined conveyor, will carry up varying slopes, which depends upon the nature of the material being handled. It is loaded in the usual way and discharges over the head pulley.

Fig. 73. Inclined or retarding conveyor for lowering material gently down slopes. May be combined with other arrangements and is used also as a boom conveyor for loading railway waggons. The conveyor and framework is hinged at the high end and can be raised and lowered to suit and also prevents breakage of fragile materials.

Fig. 74. Combination inclined and horizontal conveyor. The horizontal run can be discharged at head end or at any intermediate point by means of fixed or movable trippers, as shown in Figs. 68 and 69. The bend in carrying run can be made over a bend pulle or over a series of belt idlers as shown. This latter method is preferable.

Fig. 75. Simplest method for conveying horizontally and up an incline where belt tension is not excessive. The radius of curved section depends upon the tension of belt at curve and must be ample to prevent the belt, loaded or empty, from lifting from idlers. Maximum inclines depend upon the nature of the material being handled and the method of loading.

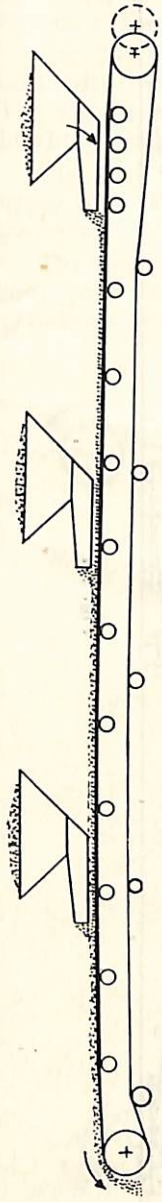


Fig. 67.

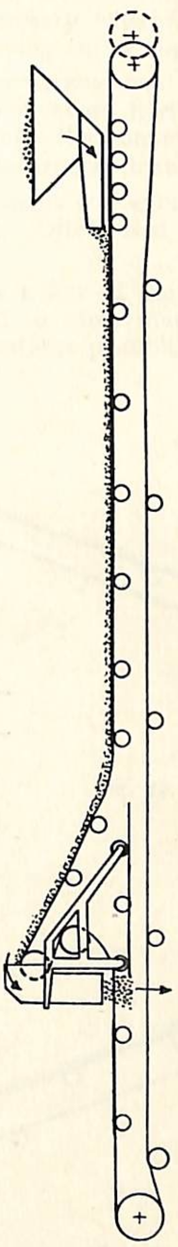


Fig. 68.

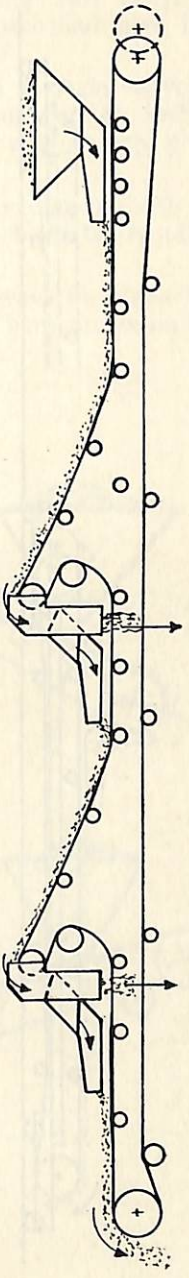


Fig. 69.

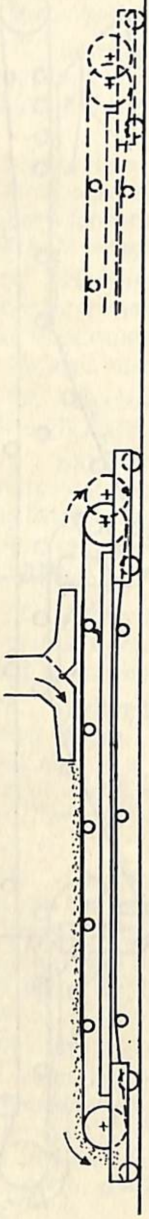


Fig. 70.

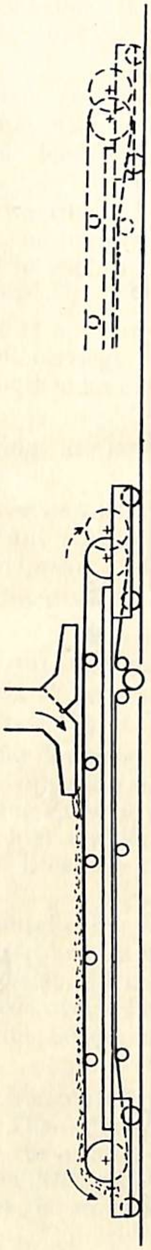


Fig. 71.

Fig. 76. Shows an arrangement combining two conveyor units and is often necessary where limited space and high belt tension make the sweeping curve impracticable.

Fig. 77. Is another arrangement in which a single conveyor unit with one drive conveyors horizontally and up an incline. The abrupt bend permits the conveyor to elevate to any given height in the shortest horizontal distance.

Fig. 78. Shows a single conveyor unit conveying on both the upper and return belt. Often used with flat belts for handling packages.

The diagrams show a few of the simpler ways in which belt conveyors can be used to carry materials. There are unlimited variations and combinations possible.

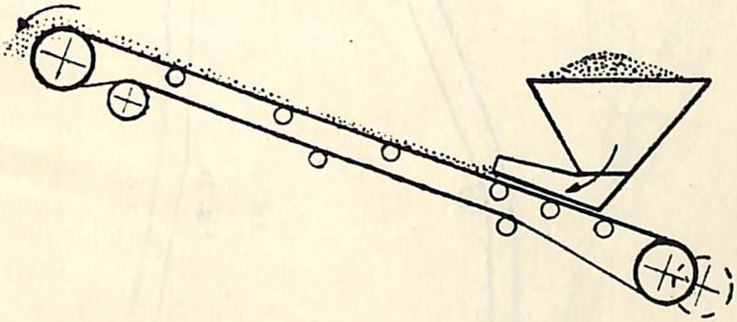


Fig. 72.

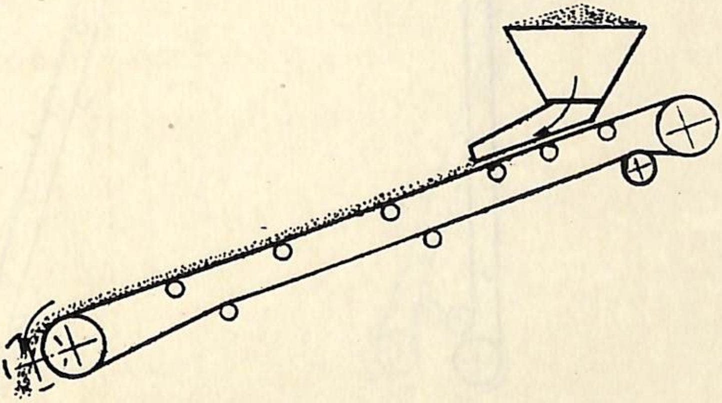


Fig. 73.

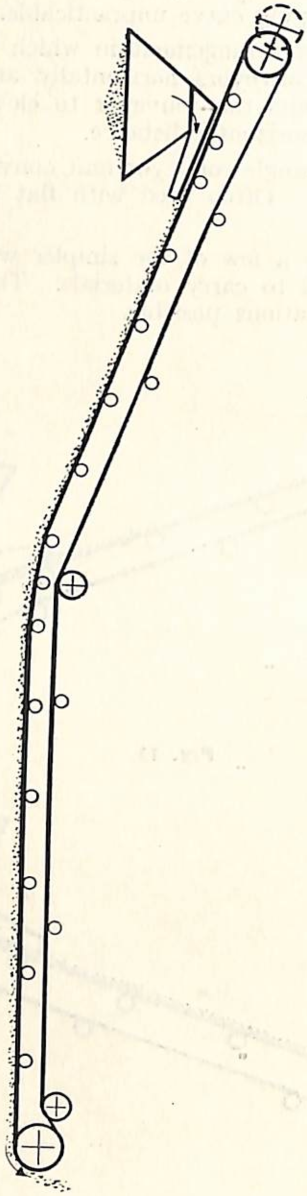


Fig. 74.

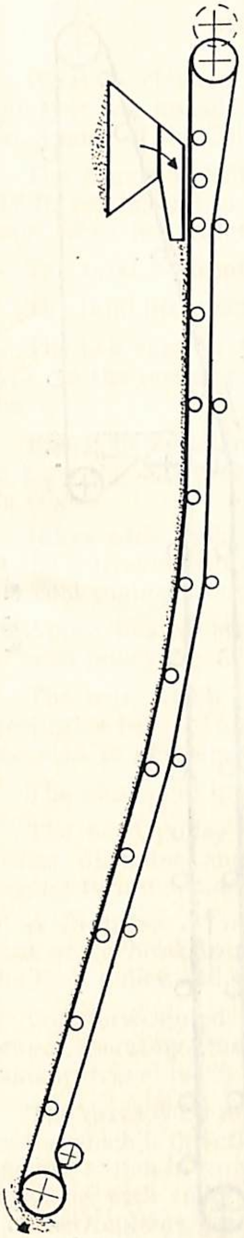


Fig. 75.

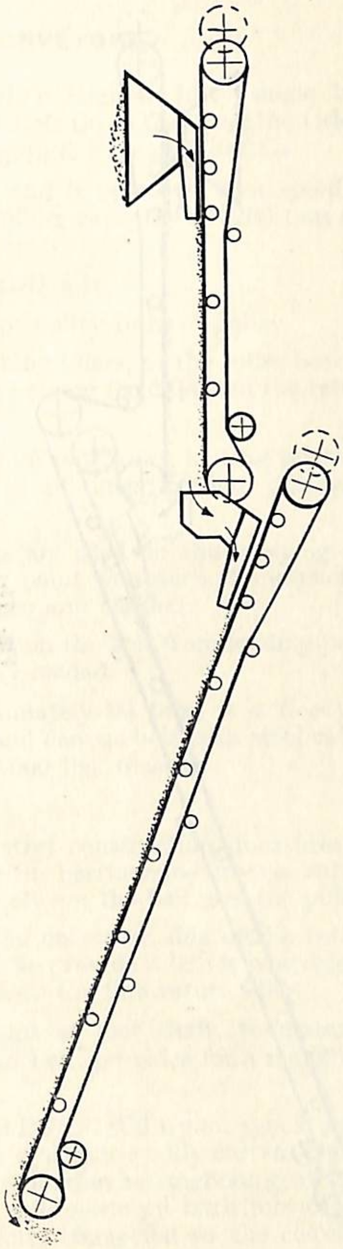


Fig. 76.

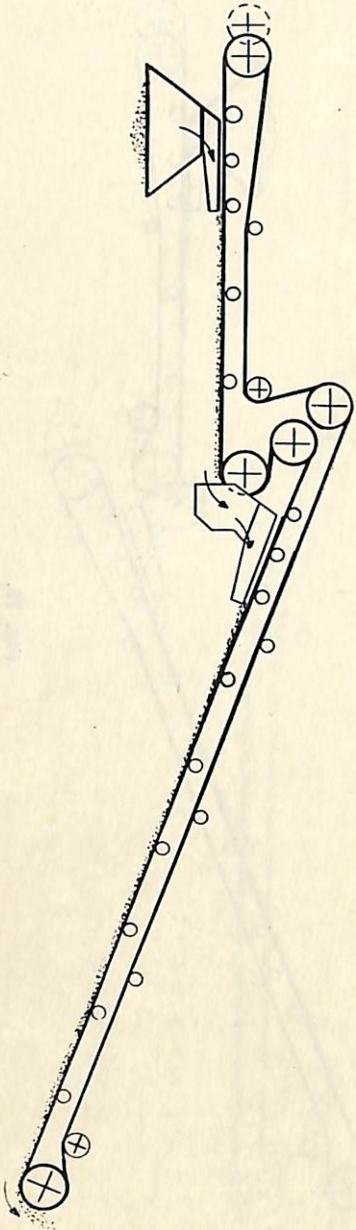


Fig. 77.

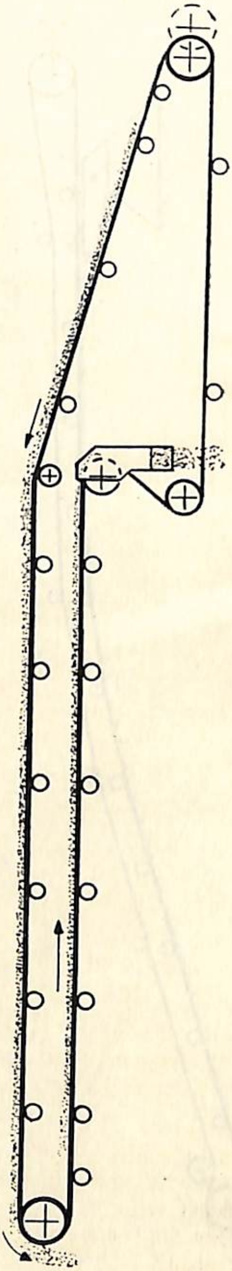


Fig. 78.

HIGH LIFT CONVEYORS.

What is claimed as the "World's Highest Lift" single belt conveyor was installed by the Link Belt Co., U.S.A., at the Orient No. 3 mine of the Chicago, Wilmington & Franklin Coal Co.

The conveyor belt is 42″ wide and is operated at a speed of 617 ft. per minute to provide a handling capacity of 1200 tons per hour, when fed uniformly at this rate.

The total horizontal length is 3167 feet.

The total lift is 862 feet from foot pulley to head pulley.

The belt runs on 3 pulley troughing idlers, of the roller bearing type, on the carrying run and roller bearing flat idlers on the return run.

Belt training idlers are installed on both runs, at long intervals to keep the conveyor belt central at all times, without damaging its edges.

Idlers with rubber covered rolls are used on the carrying run of the conveyor belt at the loading point to absorb the impact of the coal coming from a grizzly screen and crusher.

Approximately 90 tons of coal lie on the belt from loading point to head pulley when the belt is fully loaded.

The belt, which weighs approximately 80 tons, is a *Goodyear* steel cable belt. This is a rubber and canvas belt with steel cables designed to give approximately 90,000 lbs. tension.

The maximum incline was 16°.

The head pulley is of welded steel construction, measures 72 inches diameter and is fitted with herringbone-groove rubber lagging to provide added traction between the belt and the pulley.

A 48 inches diameter snub pulley on return side of the belt in back of the head drum was adopted to provide a better wrap round the head pulley and reduce the load on the first return idler.

Counterweighted gravity take-ups at foot shaft, to maintain proper operating tension of conveyor belt, provides for a maximum take-up travel of 25 feet.

The drive is by means of a 1500 B.H.P. 900 r.p.m. synchronous motor which is directly coupled to a dynamatic eddy current clutch, and the clutch in turn, to a double reduction herringbone gear drive provided with roller bearings and automatic oil bath lubrication. The herringbone gear drive is directly connected to the conveyor head shaft.

When starting the operation, the motor is brought up to speed and the dynamatic clutch then actuated to start the conveyor.

A powerful post brake, applied to an extension of the conveyor head shaft, is employed to prevent the conveyor from running backwards under load, should the current be interrupted from any cause. The conveyor belt gets an "easy" start through this drive, because the starting torque to raise the speed of the loaded belt from zero to maximum is automatically limited by the eddy current clutch.

LONG CONVEYORS.

A single two mile long 30″ wide *"Link Belt"* belt conveyor transports bituminous coal from a coal blending and washery plant to a barge and rail loading station. This conveyor operates in a rock tunnel through mountains, under woods, roads and a stream, and is claimed to be the world's longest *single* belt conveyor.

It measures 10,900 feet between the centres of foot pulley and head pulley, and employs a *Goodyear* steel cord rubber belt measuring more than four miles long.

The operating speed is 400 feet per minute and is driven by a single 200 h.p. motor, and the capacity is rated at 208 English tons per hour. It takes 36 minutes for the coal to travel from feed to discharge point.

The conveyor progresses at down grades of ·48%, then 2·076% and 5·80% to a point underneath a ravine, thereupon ascending 2·66% to the rail and river loading station.

One of the world's longest single belt conveyors system is that installed by the Stephen Adamson Co. of the U.S.A. for the Trick Colonial Mines. It is a system of 19 conveyors, totalling 4⅓ miles distance.

The belt conveyor system has a capacity of 1600 tons per hour and the record tonnage is 17,000 tons in 10 hours 33 minutes.

TYPICAL EXAMPLES.

Fig. 79 shows a large belt conveyor system installed by the Mitchell Engineering Co., Ltd., for handling coal to and from the stock pile.

Fig. 80 shows a similar plant installed by the same company taking coal to the carbonizing plant of a large London Gas Works and conveying the coke back across to a coke screening plant. Belt conveyors are used in both systems.

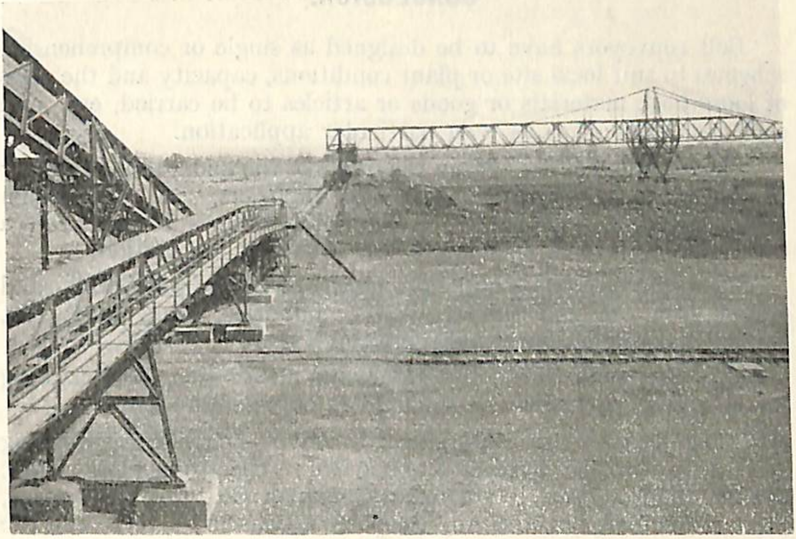


Fig. 79.

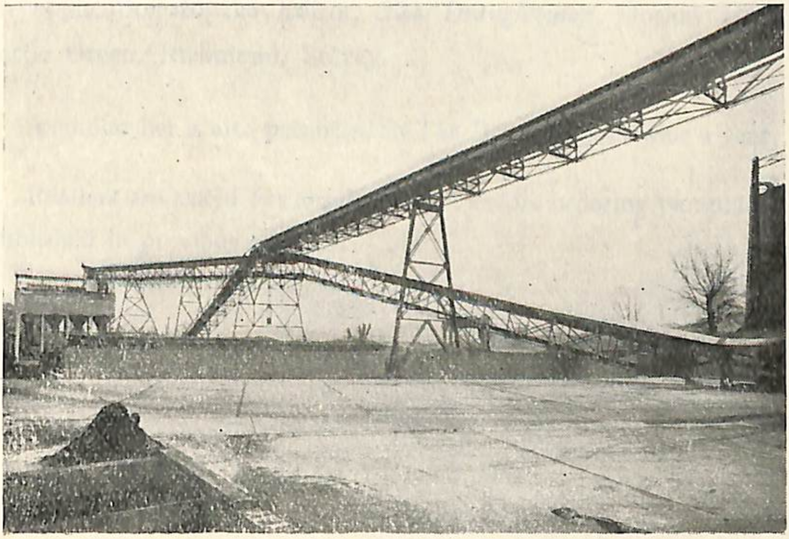


Fig. 80.

CONCLUSION.

Belt conveyors have to be designed as single or comprehensive schemes to suit local site or plant conditions, capacity and the class of loose bulk materials or goods or articles to be carried, etc., and each installation has its own particular application.

I have endeavoured, within the limits of the book, to give chiefly to draughtsmen, designers, planning engineers, etc., the design and fundamental details of the individual units rather than a descriptive article of a complete belt conveyor installation.

Such plants as are installed at power stations, gas works, sand gravel, iron ore and similar plants, foundries, grain silos, etc., employ a combination of belt conveyors.

This also applies in connection with installations for handling boxes, crates, packages and small parts on assembly lines.

I beg to offer my obligations to the :—Darrold Engineering Co., Ltd. ; British Tyre & Rubber Co. ; Electromagnets, Ltd. ; Samuel Denison, Ltd. ; Universal Conveyor Co., Ltd. ; Steelace, Ltd. ; Link Belt Co., U.S.A. ; Stephen Adamsons, U.S.A., and the many other firms who have given assistance in relation to their manufactures and loan of blocks and to whom I have tried to give full recognition in the text.

A.E.S.D. Printed Pamphlets and Other Publications in Stock.

An up-to-date list of A.E.S.D. pamphlets in stock is obtainable on application to the Editor, *The Draughtsman*, Onslow Hall, Little Green, Richmond, Surrey.

A similar list is also published in *The Draughtsman* twice a year.

Readers are asked to consult this list before ordering pamphlets published in previous sessions.

List of A.E.S.D. Data Sheets.

1. Safe Load on Machine-Cut Spur Gears.
2. Deflection of Shafts and Beams.
3. Deflection of Shafts and Beams (Instruction Sheet). } Connected.
4. Steam Radiation Heating Chart.
5. Horse-Power of Leather Belts, etc.
6. Automobile Brakes (Axle Brakes).
7. Automobile Brakes (Transmission Brakes). } Connected.
8. Capacities of Bucket Elevators.
9. Valley Angle Chart for Hoppers and Chutes.
10. Shafts up to 5½-in. diameter, subjected to Twisting and Combined Bending and Twisting.
11. Shafts, 5¾ to 26 inch diameter, subjected to Twisting and Combined Bending and Twisting.
12. Ship Derrick Booms.
13. Spiral Springs (Diameter of Round or Square Wire).
14. Spiral Springs (Compression).
15. Automobile Clutches (Cone Clutches).
16. ,, ,, (Plate Clutches).
17. Coil Friction for Belts, etc.
18. Internal Expanding Brakes. Self-Balancing Brake Shoes (Force Diagram).
19. Internal Expanding Brakes. Angular Proportions for Self-Balancing. } Connected.
20. Referred Mean Pressure Cut-Off, etc.
21. Particulars for Balata Belt Drives.
22. ⅞″ Square Duralumin Tubes as Struts.
23. 1″ ,, ,, ,, ,,
24. ¾″ Sq. Steel Tubes as Struts (30 ton yield).
25. ⅞″ ,, ,, ,, (30 ton yield).
26. 1″ ,, ,, ,, (30 ton yield).
27. ¾″ ,, ,, ,, (40 ton yield).
28. ⅞″ ,, ,, ,, (40 ton yield).
29. 1″ ,, ,, ,, (40 ton yield).
30. Moments of Inertia of Built-up Sections (Tables).
31. Moments of Inertia of Built-up Sections (Instructions and Examples). } Connected.
32. Reinforced Concrete Slabs (Line Chart).
33. Reinforced Concrete Slabs (Instructions and Examples) } Connected.
34. Capacity and Speed Chart for Troughed Band Conveyors.
35. Screw Propeller Design (Sheet 1, Diameter Chart).
36. ,, ,, ,, (Sheet 2, Pitch Chart).
37. ,, ,, ,, (Sheet 3, Notes and Examples) } Connected.
38. Open Coil Conical Springs.
39. Close Coil Conical Springs.
40. Trajectory Described by Belt Conveyors (Revised 1949),
41. Metric Equivalents.
42. Useful Conversion Factors.
43. Torsion of Non-Circular Shafts.
44. Railway Vehicles on Curves.
46. Coned Plate Development.
47. Solution of Triangles (Sheet 1, Right Angles).
48. Solution of Triangles (Sheet 2, Oblique Angles).
49. Relation between Length, Linear Movement and Angular Movement of Lever (Diagram and Notes).
50. ,, ,, ,, ,, ,, ,, ,, (Chart).
51. Helix Angle and Efficiency of Screws and Worms.
52. Approximate Radius of Gyration of Various Sections.

53. Helical Spring Graphs (Round Wire).
54. ,, ,, ,, (Round Wire). Connected.
55. ,, ,, ,, (Square Wire).
56. Relative Value of Welds to Rivets.
58. Graphs for Strength of Rectangular Flat Plates of Uniform Thickness.
59. Graphs for Deflection of Rectangular Flat Plates of Uniform Thickness.
60. Moment of Resistance of Reinforced Concrete Beams.
61. Deflection of Leaf Spring.
62. Strength of Leaf Spring.
63. Chart Showing Relationship of Various Hardness Tests.
64. Shaft Horse Power and Proportions of Worm Gear.
65. Ring with Uniform Internal Load (Tangential Strain). Connected.
66. Ring with Uniform Internal Load (Tangential Stress).
67. Hub Pressed on to Steel Shaft. (Maximum Tangential Stress at Bore of Hub).
68. Hub Pressed on to Steel Shaft. (Radial Gripping Pressure between Hub and Shaft).
69. Rotating Disc (Steel) Tangential Strain. Connected.
70. ,, ,, ,, Stress.
71. Ring with Uniform External Load, Tangential Strain. Connected.
72. ,, ,, ,, ,, ,, Stress.
73. Viscosity Temperature Chart for Converting Commercial to Absolute Viscosities. Connected.
74. Journal Friction on Bearings.
75. Ring Oil Bearings.
76. Shearing and Bearing Values for High Tensile Structural Steel Shop Rivets, in accordance with B.S.S. No. 548/1934.
78. Velocity of Flow in Pipes for a Given Delivery. Connected.
79. Delivery of Water in Pipes for a Given Head.
80. (See No. 105).
81. Involute Toothed Gearing Chart.
83. Variation of Suction Lift and Temperature for Centrifugal Pumps.
89. Curve Relating Natural Frequency and Deflection.
90. Vibration Transmissibility Curved or Elastic Suspension. Connected.
91. Instructions and Examples in the Use of Data Sheets, Nos. 89 and 90.
92. Pressure on Sides of Bunker.
93-4-5-6-7. Rolled Steel Sections.
98-99-100. Boiler Safety Valves.
102. Pressure Required for Blanking and Piercing.
103. Punch and Die Clearances for Blanking and Piercing.
104. Nomograph for Valley Angles of Hoppers and Chutes.
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106. Compound Cylinder (Similar Material) Radial Pressure of Common Diameter (D1).

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